Once There Was a Republic

THE NATION had been "blessed with victory and peace" but soon there came years in which unemployment and hunger stalked the land. It was the backwash of the terrific war that had been fought; the pendulum was swinging back. In their discouragement the people forgot patriotism, self-sacrifice, high ideals, the virtues of hard work and spiritual values. Thinking was smothered by mental confusion.

Out of the chaos there came a promising leader. Wilingly and enthusiastically, the masses followed him. He curbed the greed and selfishness of the rich and of the leaders of industry. To the fullest extent of his power to do so, he forced employers to give more work and more pay. But the masses were impatient, and unscrupulous leaders of labor seized the opportunity. Well-meaning and ardent social welfare workers unwittingly fanned the smoldering discontent. Impractical experiments were tried; more legislation was passed to curb the employers. Results were disastrous. Production of wealth steadily decreased; strikes followed each other in rapid succession; industry became demoralized, all but stagnant.

War clouds gathered. Another nation was threateningly preparing for conquest. Immediately the Republic contracted for huge orders of armaments and fighting equipment. The young men were called for military training. Millions were put under arms. But something was lacking. Although the government succeeded in controlling strikes to a great extent, there was little or no enthusiasm on the part of the workers. Although every attempt was made to build up a feeling of patriotism in the hearts of the soldiers, they reacted without spirit. And the masses of the people did nothing. The nation was conquered.

Well-meaning Premier Blum who had been in power from 1934 to 1937 had done much to improve, at least temporarily, the conditions of the French working man. But many of his experiments served only to becloud the fact that the virtues of a nation are derived from the hard work, clean living and high morals of its inhabitants. When citizens forget to "praise the power that has made and preserved them a nation," that nation is doomed. In the case of France, it has been summed up in this brief sentence—"The soul of France was sick."

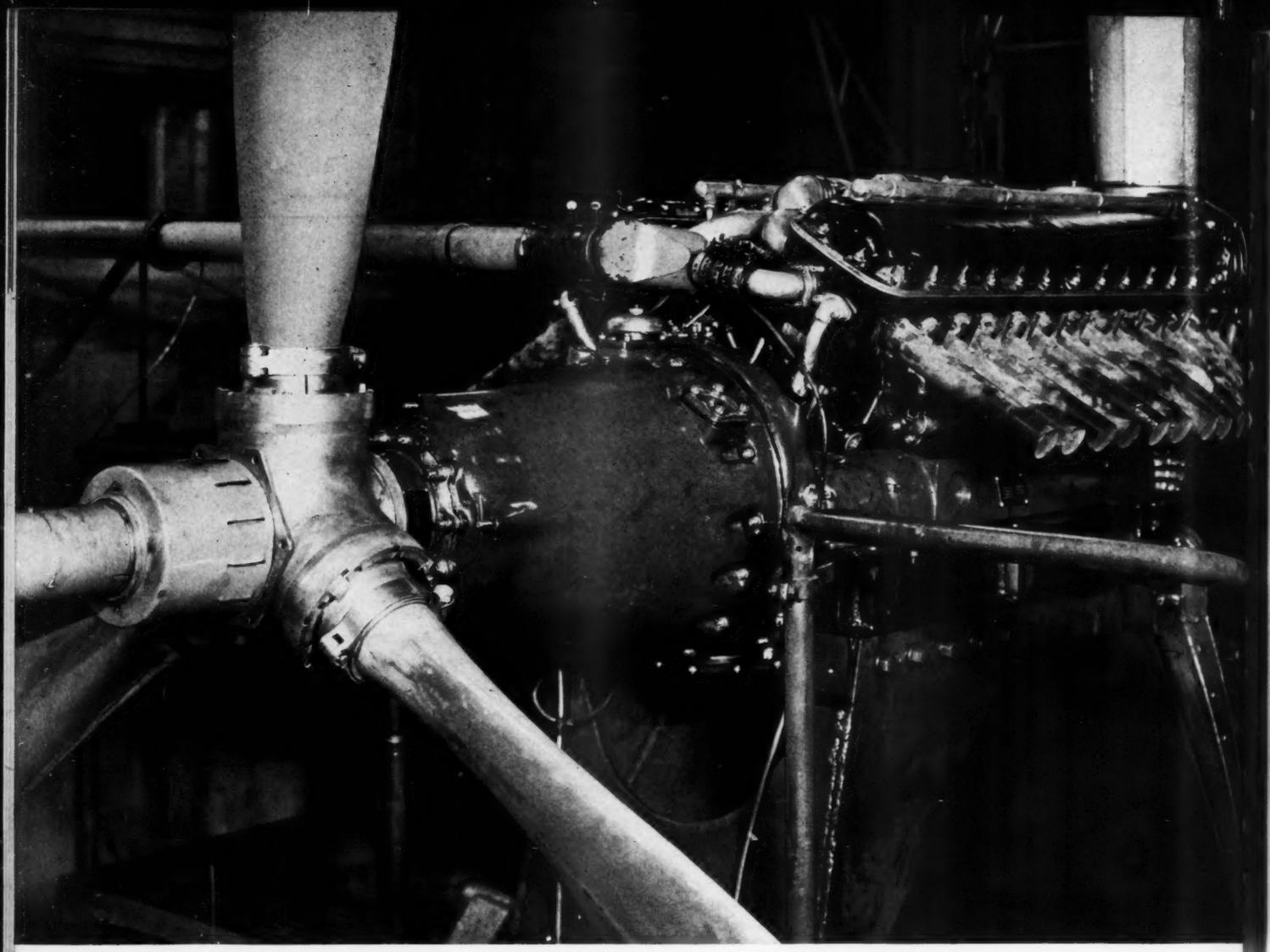
For almost a year it has been obvious that in the all too-near future we will be compelled to defend our liberties and our institutions. But still there is bickering over pennies, and men strike, sitdown, or otherwise stop work for little or no reason. Impatience over wages and working conditions overpowers the spirit to "stand between our loved homes and war's desolution." And masses of the people are being lulled into a false sense of security by self-styled experts who on their own imagined premises and assumptions prove that Hitler cannot conquer America.

Other than national spirit, there is little that is comparable between France and the United States. We are blessed with a leader who ranks with our greatest. But vainly does he wait for the voice of his people to be heard.

What has happened to America? Is our soul still sick? Why do not our great "service clubs," Rotarians, Kiwanis, Lions and others, relight the torch of freedom and give vent to the pent-up feelings that are swelling in the hearts of true Americans? Why do the officers of municipalities allow draftees to leave for camps with no more attention than is given the daily commuter? Where are the parades, the roll of drums, the patriotic exercises, the salute to the flag, the singing of the Star Spangled Banner?

Wake up America! We must return quickly to the spirit of our pioneers who, to establish and preserve their freedom of life and worship, conquered the wilderness, fought the Indians, endured the elements and praised God.

Or shall it also be written of these great United States —once there was a republic but its soul became sick?



THE ALLISON ENGINE

A record of its development from an idea to mass production

G. F. NORDENHOLT

MOST new scientific and engineering developments in their early stages become the targets of irresponsible criticism. The laymen on the banks of the Hudson knew nothing about either the steam engine or navigation, but they lustily derided Fulton and his steamboat. The people who knew least about engines and aerodynamics laughed heartily over the attempts of the Wright brothers to build a flying machine. The caustic criticisms that have been directed against the Allison Division of General Motors and the reports of failures of the Allison engine both in performance and production, are in the same general category. For the most part they can be credited to irresponsible statements relayed by uninformed reporters to a daily press that publishes items without bothering to check the supposed facts. An accurate account of the development of the Allison engine has never been given. General Motors obviously preferred to ignore the irresponsible criticisms and distorted reports. After all, such stories did not affect the development program one iota, so there was no reason for General Motors to pay the least attention to the disparaging gossip concerning the Allison engine. But on the other hand, in justice to the men who perfected the world's most powerful liquid-cooled airplane engine and put it into mass production in a remarkably short time, the story should be told.

Actually, Jim Allison did not design the Allison engine. He was neither an engineer nor a mechanic; he was a wealthy sportsman, a keen business man and an amazing character. To Norman H. Gilman, Jim Allison's protegé, should go the bulk of the credit for having initiated the design of the world's most powerful liquid-cooled airplane engine. But Gilman always in-

sists that if it had not been for Jim Allison there would be no Allison engine today, and that also is true.

Be that as it may, the events which led to the design of the Allison engine can be said to have had their beginnings at the Indiana State Fair in Indianapolis on Labor Day, 1905. It was there that Jim Allison and his partner Carl Fisher, owners of the Prest-O-Lite Company, gathered with their many friends of the automobile industry of that day, to watch Barney Oldfield drive his Green Dragon racer to win. Those were the days when Arthur Newby was principal owner of the then famous National Motor Car and Vehicle Corporation of Indianapolis, when Frank Wheeler was head of Wheeler-Schebler Carburetor Company and Harry Stutz was building automobiles. All were friends of Jim Allison, and all were in close accord.

Besides the colorful daredevil Barney Oldfield, the top-ranking racing car drivers of that day included Charlie Merz, during the World War a major in the Air Service in France, now president of Merz Engineering Company of Indianapolis. Another one was Eddie Rickenbacker, later a war ace, now president of Eastern Airlines and of the Indianapolis Motor Speedway.

Johnny Aiken was the ranking driver of National Cars. Supervising the servicing of the National Cars entered in the races, was a 26 year old Yankee-born mechanic, at that time assistant superintendent of the National Motor Car and Vehicle Corporation shops. His name was Norman H. Gilman.

Sportsman Jim Allison, highly respected for his keen business judgment, joined with Carl Fisher, Arthur Newby and Frank Wheeler to build the famous Indianapolis Speedway. Allison's sporting nature and love for automobiles led him to sponsor three automobile racing teams. This required a shop for building and servicing cars and in 1916 he built such a shop close to the Indianapolis Speedway. He needed a superintendent to run the shop. Through Johnny Aiken, who had become Jim Allison's ranking driver, he heard that Norman Gilman was looking for a job, National Motors being then on the way out. Thus it came about that on January 1, 1917, Norman H. Gilman became superintendent, and chief engineer of Jim Allison's Indianapolis Speedway Team's 20-man shop. The shop equipment and personnel reflected the spirit of Jim Allison. The machine tools were the best that could be purchased; the mechanics the best that could be had.

Then came the war. Jim Allison called off automobile racing for the duration of the war and ordered Norman Gilman to forget about automobiles and get war jobs. "Take any job you like, especially the ones the other fellow can't do; take anything that will help the other fellow to get his production rolling." The Nordyke-Marmon Company had an order for Hall Scott and Liberty aircraft engines; Allison's shop built the tools, jigs and fixtures. Two Liberty 12-cylinder Vee engines were needed as production models by other companies, and Norman Gilman and his mechanics built them. These were no juicy war contracts; it was all experimental work on which profits were small and losses big.

The 20-man shop expanded to 50 and then finally 100 mechanics. Only difficult jobs were thrown in their laps. Production models of superchargers, trucks, whippet tanks, and high-speed tractors were developed. To coordinate the maze of drawings involved and to keep them abreast of the design changes, a force of 150 designers and draftsmen was organized. Under Norman H. Gilman the shop of the Allison Speedway Team Company produced design models with a speed and perfection that made them famous.

In 1917 the name of the company was changed to the Allison Experimental Company. Work was scarce after the war but the shop was never closed. Then Jim Allison had Gilman prepare a car for the 1919 500-mile International Sweepstakes. The car won the race. After that Jim Allison quit racing and sold all his cars to Ruth Law the aviatrix who used them in her "plane-to-auto" stunting on barnstorming exhibitions. In the fall of 1919 the name of the company was changed to the Allison Engineering Company.

About 1927 the Navy and the Army started to give the Allison shop special jobs to do. These included rebuilding Liberty engines. One of them was rebuilt to use 20 percent benzol mixed with the aviation fuel. The idea was to thereby increase the horsepower output of the engine from 450 to 570 horsepower. Norman Gilman rebuilt the engine which ran under test for 31 hours and then broke. The connecting rod bearings had failed.

This incident had a far reaching effect. Norman Gilman had become intensely interested in the possibilities of a liquid-cooled airplane engine. The air-cooled radial engine appeared to him to have definite limitations. He reasoned that 500 horsepower would be about as big as it would be possible to build an air-cooled engine. In this he was wrong. But there was also the other factor that a radial air-cooled

James A. Allison — Keen business man and wealthy sportsman. He told his protegé Norman Gilman, "We are not running this shop for profit. Your job is to build the best engines possible." He did not live to see the Allison airplane engine

Norman H. Gilman — He pioneered the development of the Allison engine, but insists that "If it had not been for Jim Allison, there would be no Allison engine today." He was the first president of the Allison Division of General Motors

Jim Allison's shop near the Indianapolis Speedway Track as it appeared in 1916. It is now a part of Plant No. 1, Allison Division—General Motors Corporation. Nearby are Plants 2 and 3, one-story buildings, totaling more than 25 acres of floor space

engine mounted directly in the wind stream, presented a large flat surface that imposed a great drag on the speed of the plane. A liquid-cooled engine mounted inside the plane would offer no added wind resistance or drag. Norman Gilman determined to build a 1,000 hp. liquid-cooled airplane engine.

Because the bronze-backed babbitt bearings had failed on the Liberty engine, Gilman decided that the first thing to do was to develop a type of bearing that could withstand greater loads. This is what led to his invention of the steel-backed, leaded-bronze bearing, a type of bearing still used almost exclusively in all airplane engines. The thin bronze welded to the steel does not pound out as did the babbitt; the more rigid steel backing hold the bearing in its circular shape. In later years it was the profit from the manufacture and sale of the patented Gilman bearings that paid some of the early development costs of the Allison engine.

It was at about this time that Norman Gilman needed an assistant chief engineer and hired John L. Goldthwaite to fill the vacancy. Goldthwaite became highly expert in the solution at vibrations problems and it was he who in later years was mainly responsible for the development of the ingenious means at vibration damping now used on the Allison engine.

But Gilman was unable to proceed immediately with his ideas of a 1,000 hp. engine. Other jobs intervened, and then on August 3, 1928, Jim Allison died of pneumonia. The executors of the estate wanted to get rid of the shop. General Motors bought it and for several years continued to operate the shop under the name of Allison Engineering Company. Gilman was made president.

Gilman had not given up his idea of a 1,000-hp. liquid-cooled airplane engine. But he had a hard time selling the idea to others. The experience of the Curtiss Airplane and Engine Company with the Curtiss V-12 engine was discouraging others from attempting to develop a high horsepower liquid-cooled engine. The Curtiss V-12 Conquerer was a water-cooled engine which at 20,000 ft. elevation necessarily had to operate with a radiator water temperature less than 160 deg. F. Curtiss engineers got the idea of using ethylene glycol instead of water in the radiator and allowing the radiator temperatures to run up to 300 deg. It was expected they would thereby be able to step up appreciably the power output from its standard rating of 550 hp. But, although the Curtiss V-12 was an excellent 550 hp. water-cooled engine, it was a complete failure when it was attempted to boost the horsepower by using ethylene glycol for cooling.

But in spite of the failure of that Curtiss engine, Gilman felt certain that he could develop a successful design of a 1,000-hp. liquid cooled engine. He drew up sketches and made a rough design and showed them to Army engineers with the suggestion that the engine be used in a bomber. But the Army was not interested. He showed his designs to Navy engineers and they became interested. In November 1931, the first liquid-cooled airplane engine of 750 hp., 2,400 r.p.m. was ordered by the Navy to be built by Allison Engineering. This engine was designed and built, and delivered to the Navy on March 12, 1932. In September of the same year the 50-hr. development test was completed by the Navy and the engine was officially accepted. During all of these tests only insignificant details in the design were changed.

The Navy also asked Norman Gilman to develop similar engines for the airships Akron and Macon which were powered by German Mayback engines. For obvious reasons the Navy did not like the idea of being compelled to use German engines in United States airships. Similar to the first engine, the airship engines were to develop 650 hp., run at 2,400 r.p.m., but were to be able to run in either direction of rotation. Allison accepted the Navy contract to build three of these engines and the first one was completed in September 1933 and passed its type test in September 1934. The other two engines were built for the Macon and the day they were being packed for shipment the news came that the Macon had gone down in the Pacific Ocean. For the time being this stopped all further work on the development of liquid-cooled engines for the Navy.

While the airship engines were being built for the Navy, the Army Air Corps became interested in the possibilities of a liquid-cooled engine and Allison received from them an order to develop a 750 hp. engine operating at 2,600 r.p.m. General Motors agreed to back Gilman in the acceptance of this order. The engine was designed and delivered to the Air Corps in June 1933. It ran successfully on the preliminary brake test for 50 hours without developing any trouble.

The idea was to repeat the tests, but in each successive test run the engine at increased horsepower output. It was expected that by replacing the parts that might fail at the higher horsepower outputs with redesigned parts of greater strength, a successful 1,000 hp. engine would be achieved.

It was also in the summer of 1933 that there came to Allison Engineering a young man named Ronald M. Hazen who had been assistant chief engineer for Ranger Aircraft Engine, Inc., a

division of Fairchild Engine and Airplane Corp. Hazen had done notable work in the development of the Ranger engine, but the depth of the depression had left him without a job. Norman Gilman hired Hazen and put him to work on special problems relating to designs of engines. In order to work out some of these special projects more effectively, Hazen was transferred to the General Motors Research Laboratories in Detroit.

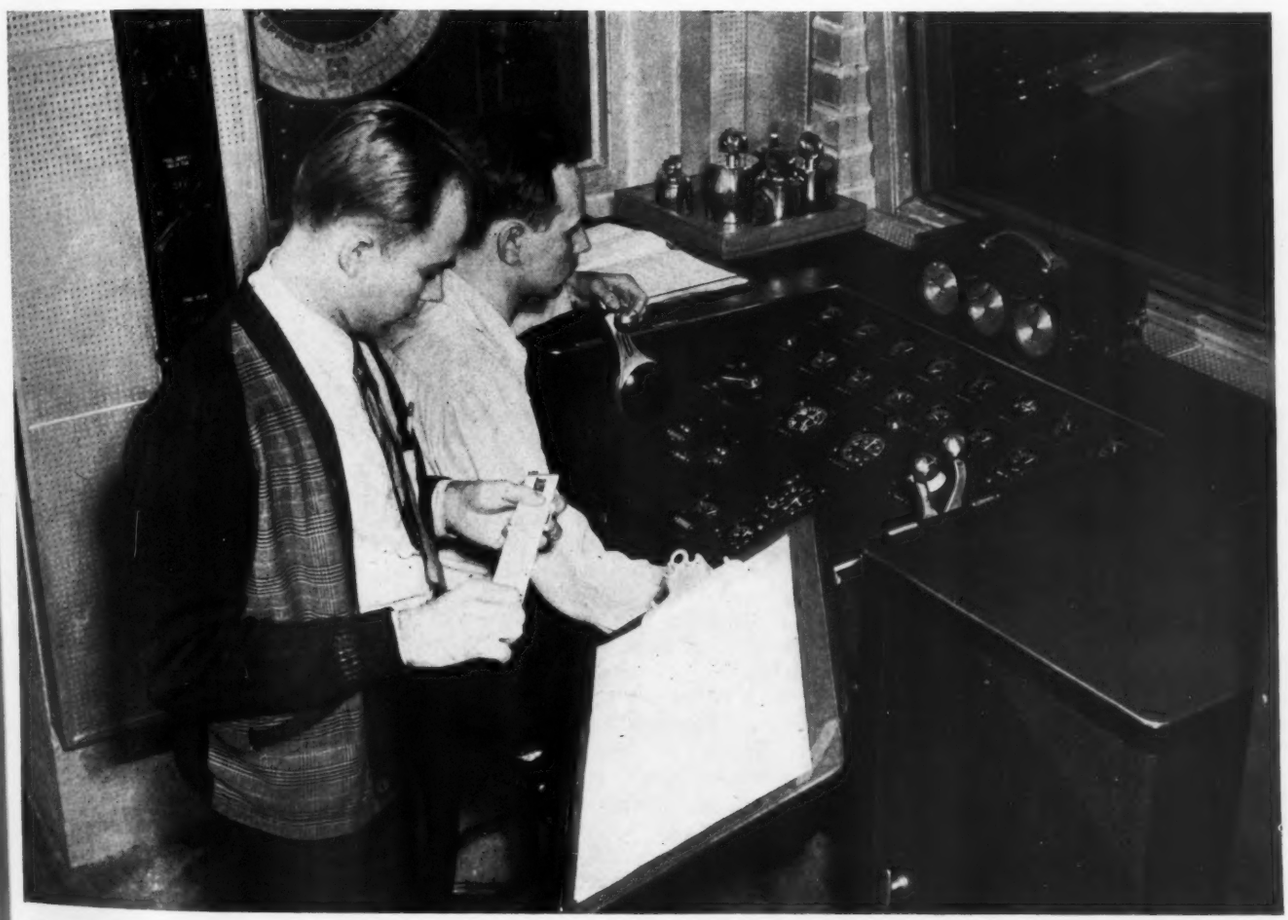
As mentioned, the first model of the 1,000 hp. engine passed its 50-hr. preliminary test in June 1935 with flying colors. But when the tests were continued at increasingly higher horsepower outputs, everything went haywire. Vibrations twisted off the crankshaft, crankcases cracked, valves burned out, bearings failed, everything failed. For a year and nine months, from June 1934 to March 1936, Gilman and his engineers and mechanics worked like Trojans but it seemed that everything went against them. Gilman finally decided to completely redesign the engine and looked around for an engineer to supervise the job. He thought of Ronald Hazen, Hazen who had put the Ranger engine across, and decided that he was the man to perfect the V-1710, the 1,000 hp. Allison engine.

In March 1936 Ronald Hazen was made chief engineer of the Allison Division of General Motors. The way Gilman states it, "Since the very beginning of Allison Engineering I always held the title of chief engineer but when I decided to get Ronald Hazen back to head up engineering at Allison, I felt that here was the man who should succeed me as chief engineer." Gilman's judgment proved to be sound.

J. L. Goldthwaite became Ronald Hazen's right-hand man. Hazen's first decision as chief engineer was to modify entirely the existing design of the 1,000 hp. engine. On March 7, 1936 Gilman, Hazen, Goldthwaite and several other engineers started a complete redesign of the engine. Between March 7 and June 13, 1936 the new engine was designed and a model was built and delivered to the Army Air Corps, all in three months and a week.

One afternoon in the summer of 1936 Norman H. Gilman was sitting in his office when he received a message from Wright Field, Dayton, Ohio. The engine had successfully run 140 hr. and 30 min. As he read this message, Gilman felt sure that his great ambition had been accomplished, that at last his dream of a 1,000 hp. liquid-cooled airplane engine was fulfilled. He had worked hard, he was past middle age, and the terrific pace was beginning to tell. In the course of his highly successful career he had saved enough money to assure him a comfortable retirement. There was no question in his mind that the test engine would keep on running for another 9½ hr., which would complete the 150 hr. acceptance type test and thereby establish itself as the first 1,000 hp. airplane engine to pass that test. The time to retire had arrived. He called in his secretary and dictated his resignation which he put in the drawer of his desk for signing the next morning. But the next morning he learned that shortly after the engine had been started again to complete the last 9½ hours of its acceptance test, the cylinder head had cracked. Gilman tore up his resignation. There was still work to be done.

Ronald Hazen undertook a detailed



From inside the sound-proof room test engineers control the engine and through the triple-glass window watch it running. Instruments indicate all pressures and temperatures, fuel and oil supply and thus the performance of the engine is determined

study to find out what had caused the cylinder head to crack. An exhaustive study was made not only of the design of the cylinder head but also the manifold. Out of this there developed a manifold of improved design and a cylinder head that withstood all tests at continued high duty. After many tests in the Allison laboratory, a new test engine was built and delivered to Wright Field. This time Norman Gilman felt so sure of the success of this redesigned 1,000 hp. engine that he resigned in December 1936 and retired. He was succeeded by O. T. Kreusser, who in September of that year had been transferred to Allison from General Motors Research.

Although it sounds simple to redesign a cylinder head and manifold, it was not until March 23, 1937 that the redesigned engine was completed and delivered to Wright Field. And Norman Gilman's confidence was fully justified. Late in April, 1937, the U. S. Army Air Corps at Wright Field completed the type test on the V-1710 Allison Engine. It passed the test and during that same year a Curtiss single engine XP-37 Pursuit plane and a Bell Twin Engine XFM-1 Fighter were flown with 1,000 hp. Allison engines. They performed so well that the Air Corps decided to get more of the engines to enable them to make more extensive service tests.

In the spring of 1939 the Curtiss XP-40 airplane powered by an Allison V-1710 engine won the Pursuit Competition, a competition conducted by the U. S. Army Air Corps to determine the relative superiority of various types of planes and engines. The Allison Division—General Motors Corporation was clearly in the picture as a production source for military aircraft engines particularly adapted to the higher speed airplanes needed to provide the United States with planes as fast or faster than Europe's fastest.

It was at this point in the history of the Allison engine that the tremendous power of the huge General Motors Corporation began to make itself felt. General Motors officials as early as the spring of 1939 recognized the rapid development of war-like conditions in Europe. So sure were they that there would be a tremendous need for aircraft engines that in May 1939 ground was broken for a factory and office building at the Allison plant in Indianapolis, with a total floor space of 360,000 sq. ft. Machine equipment and tools were immediately ordered for this expanded factory. And one month later Allison Division—General Motors Corporation received a contract from the Army Air Corps for engines. Production was started in February 1940.

As was to be expected, when the Allison engines began to be installed in the U. S. Army Air Corps airplanes, various troubles developed. For example, although the engine installed in the Curtiss XP-40 plane which won the 1939 Pursuit competition functioned perfectly, when the same engine was mounted in another Curtiss XP-40 the crankshaft bearings failed in a short time. Irresponsible newspaper reporters, blamed the failure on the engine. But the fact was that the failure was caused by changing the position of the oil line running from the lubricating oil tank behind the pilot to the engine in front of him. In the first plane, this line was carried over the wing, while in the second plane the line was carried under the wing and larger tubing was used. The new position of the lubricating oil line permitted air to get in the oil line when the plane was put through quick maneuvers. And after the plane leveled off, the volume of air in the larger oil line was so great that the oil pump could not pump it out quickly enough to give the bearings the needed oil before they burned out. Yet no one could have foreseen this difficulty. And a simple change in the design of the oil tank cured the trouble.

Similarly in another installation the breather pipe in the crankcase was shifted a distance of about 8 in. in order to make room for a part of the plane structure. No one gave it much thought because it was such a minor change. But it happened that the new position of the breather pipe was such that it permitted cold air to blow into the crankcase directly upon the thrust bearing. Deprived of oil and with a stream of moist air constantly impinging upon it, the bearing began to rust and soon failed. Here again the design of the engine was blamed and Allison was held up to ridicule by an uninformed press.

On a later occasion the Army Air Corps decided to make changes in the ratings of Allison engines. Whereas the Allison V-1710 engine with a supercharger develops 1,200 hp. at 20,000 ft. elevation, without a supercharger this same engine develops only 900 hp., which is entirely natural because of the reduced weight of air and fuel fed to the engine cylinder in the rare atmosphere at 20,000 ft. Apparently the Army Air Corps decided that they would specify the Allison V-1710 engine without supercharger as a 900 hp. engine. Irresponsible newspaper men made capital of this with a bold heading to the effect that "because of engineering difficulties" the Army Air Corps had reduced the horsepower rating of the Allison engine from 1,200 hp. to 900 hp., a ridiculous statement in the light of the facts.

As all engineers know, in every new development there are bound to be some weaknesses. Considering the culties and complex problems in the design of a 1,000 hp. engine of minimum weight and an entirely new cooling system and the problems of coordinating the engine and plane characteristics, the speed with which the Allison engine reached its present high degree of perfection is a remarkable engineering performance.

When in the spring of 1939 General Motors Corporation definitely committed itself to the production of airplane engines, it immediately concentrated the tremendous powers of its far-flung organization on the task of putting the Allison Division on a mass production basis in the shortest possible time. Under O. T. Kreusser, general manager of the Allison Division, production facilities and personnel were expanded rapidly. Many of General Motors most experienced production executives were transferred to the Allison Division. W. H. McCoy of the General Motors Research Division, Detroit, was assigned the responsibility for tooling and planning. W. G. Guthrie, formerly works manager of the Opal Division of General Motors became works manager in June 1940. B. Conway, general master mechanic at the Pontiac Division arrived in March 1940, to become superintendent of Plant No. 3. He is now general superintendent of production at Allison. C. M. Jessup, production manager of the Delco-Remy Division became manager of the Materials Control Department of Allison. R. E. Smith, formerly inspection superintendent of A. C. Spark Plug became chief inspector. C. A. Riddell, formerly manufacturing expense analyst of General Motors Overseas Division in New York was appointed standards superintendent, and F. C. Tykle, formerly assistant superintendent of Plant No. 2 of Delco-Remy became plant engineer at Allison. T. S. McCrae was promoted from field engineer to assistant chief engineer, and R. E. Lynch, formerly product engineer at the Diesel Engine Division of General Motors became assistant to Ronald Hazen. Owne Macker and Jack Gorden of Cadillac, John Dolza of Buick, and many other highly experienced design and production engineers in the vast General Motors Corporation came in from all directions, key men around whom was built with amazing speed and efficiency an organization of thousands of em-

ployees. Only a large corporation like General Motors could have done it; only a far flung organization that had within itself the talent required. In August 1940, further expansion and more intensified production was demanded. F. C. Kroeger, general manager of Delco-Remy Division and considered one of the most successful manufacturing managers of General Motors, was appointed general manager of the Allison Division. O. T. Kreusser continued as president of the Allison Company and also became director of service and training.

Besides the talent assembled from the various General Motors Divisions, many employees of the original Allison Company have continued in the service. C. C. Anderson who was assistant superintendent is now superintendent of the experimental machine shop of the engineering department. George Zink, who took a large part in the development of the bearings is now manager of Plant No. 1 which is devoted entirely to the manufacture of these bearings.

In addition to speeding production by rapidly building the Allison organization, further speed was gained by assigning various General Motors Divisions to build parts. Cadillac was given the job of turning out connecting rods and crankshafts; Delco-Remy furnishes small parts and accessories.

To Norman H. Gilman who in 1940 was voted by his associates to be honored by the American Manufacturers Association as a "Modern Pioneer on the Frontiers of American Industry" goes the credit for having conceived and developed the design of the Allison engine. But the whole undertaking was vastly greater than a one man job. As Norman Gilman pointed, "I cannot tell how I got any particular idea. You talk to one man and that starts you off on another track. Bye and bye your thoughts start crystalizing and finally you get a definite idea. But I do know that if it had not been for Jim Allison I would never have built that engine."

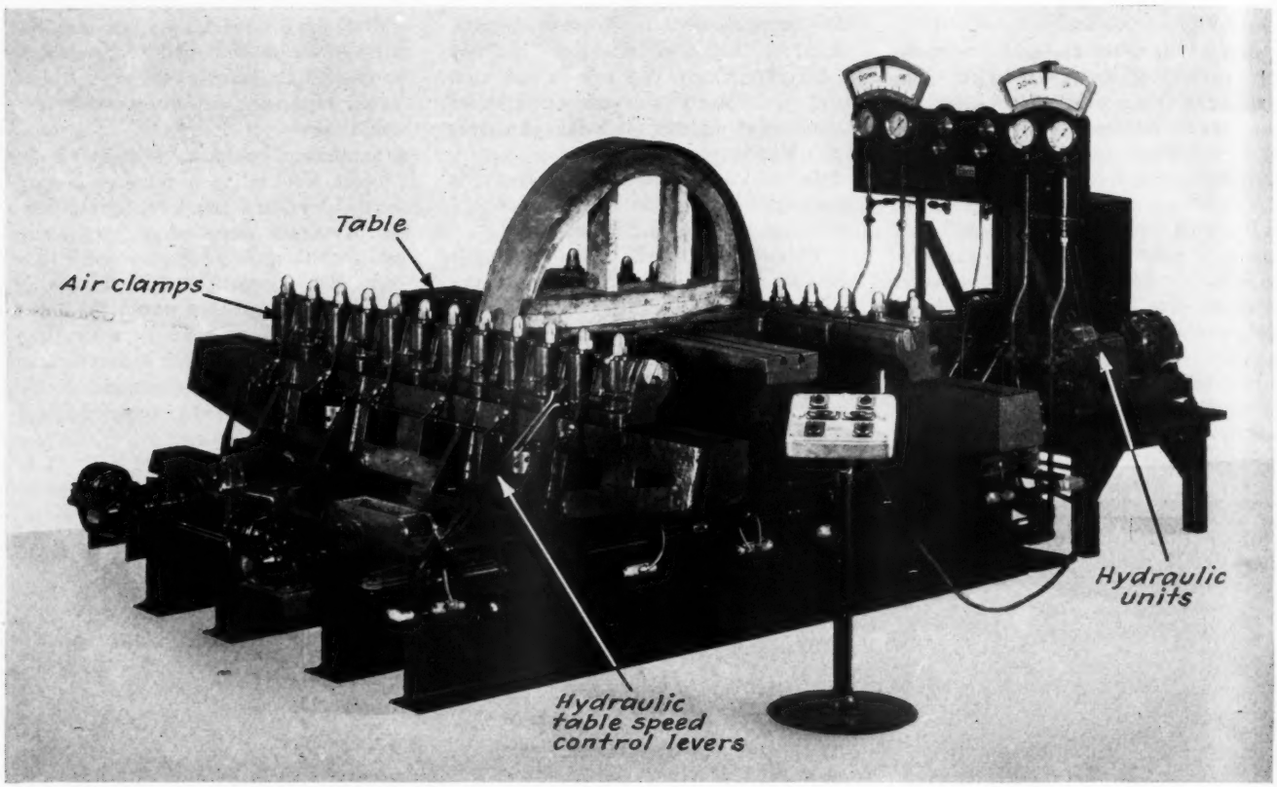
That is undoubtedly true. And so it is that into the development of the Allison engine went the spirit of Jim Allison and his associates, the men with whom Norman Gilman came in contact. Many of them were the outstanding men of their time, men who were laying the groundwork for the world's greatest industry—the American automobile. Arthur Newby of National Motor Car and Vehicle Corporation, optimist Carl Fisher whose eyes sparkled like diamonds when he got a big idea, Frank Wheeler of Wheeler Schebler Carburetor Company, daredevil Barney Oldfield and colorful Eddie Rickenbacker, Johnny Aiken, Joe Dawson, these and many others set the tempo of the period when the Allison engine idea was born and undoubtedly made their imprint on the mind of Norman Gilman.

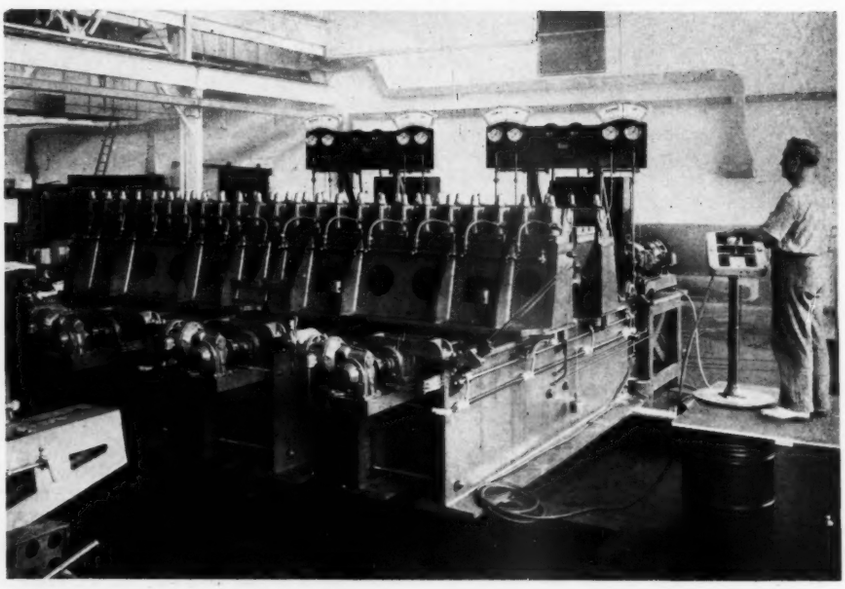
But when the Allison engine idea had grown into a definite production project, it was only the tremendous power of a far flung organization that could have put it across. No small organization could have financed the project, could have furnished the research facilities required, would have had available the kind of engineering and manufacturing talent needed. The men of General Motors can well be proud of the job they have done. And to Norman H. Gilman goes the happiness, satisfaction and honor of having pioneered the creation of the world's finest and most powerful liquid-cooled airplane engine.

Allison-powered Curtiss P-40 advanced pursuit planes are unusually fast, highly maneuverable and well armed. They are being produced in large numbers for the U. S. Army Air Corps by the Curtiss-Wright Corporation at its Buffalo, N. Y. plant

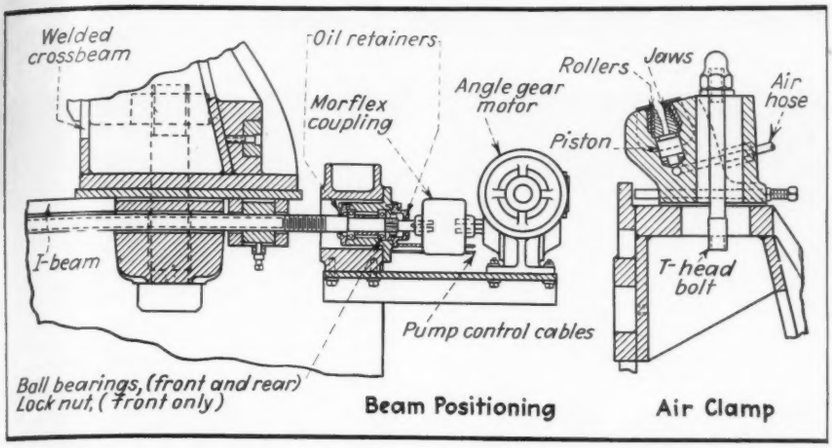
Modern Designs

Sheet Metal Stretching Press

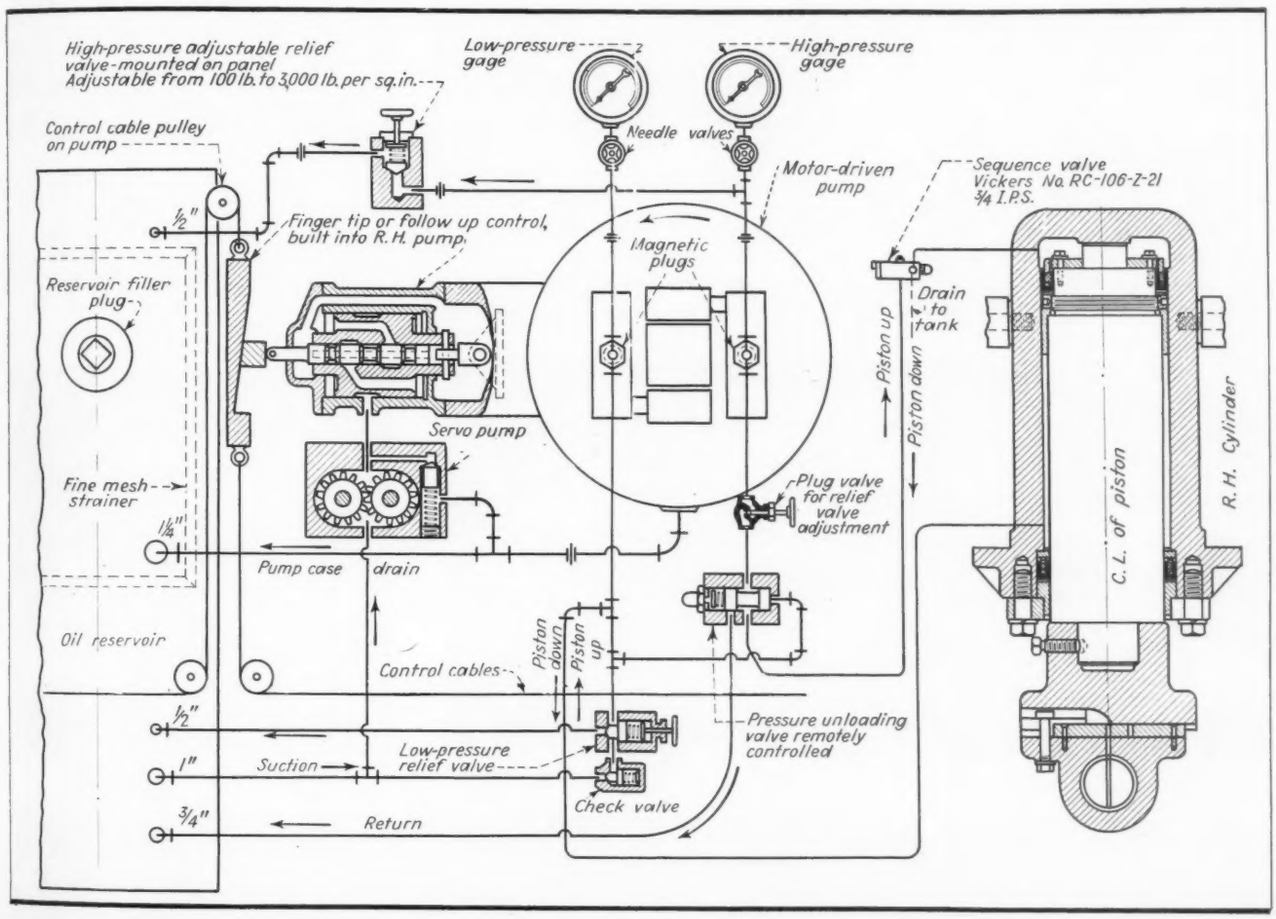




Hydraulically-operated presses for stretching metal sheet into double curvatures have been developed by the Engineering & Research Corporation. The presses were developed primarily for the aircraft industry to be used for forming wing and fuselage coverings. Sizes of presses range from a 75-ton single cylinder to the 300-ton double-cylinder machine shown above. The metal sheet is held at two opposite edges by air clamps mounted on heavy welded beams which can be set parallel to each other (24 to 80 in. apart) or to any included angle up to 20 deg. Center table is moved vertically by two hydraulic cylinders in which the pressure can be varied from 0 to 3,000 lb. per sq. in. and the speed of piston travel from 0 to 10 in. per min. depending on load. Combination of machines is possible for greater capacity. The two 300-ton machines at left are coupled.



Cross-beams are moved by 1½-hp. special high torque geared motors mounted at the end of the supporting I-beam. Cross-section at left shows motor driven screw and ball bearing mountings. Jaws of clamps are air actuated, and are forced against sheet by angular position of plates in back of rollers as jaws move upward. Screw bearing housings are packed in a high-grade acid-free ball bearing grease of No. 2 consistency, Texas Company Starfax No. 2, or equivalent. Screws are lubricated by hand with heavy oil. Table pins and table guide are lubricated through Zerk fittings.

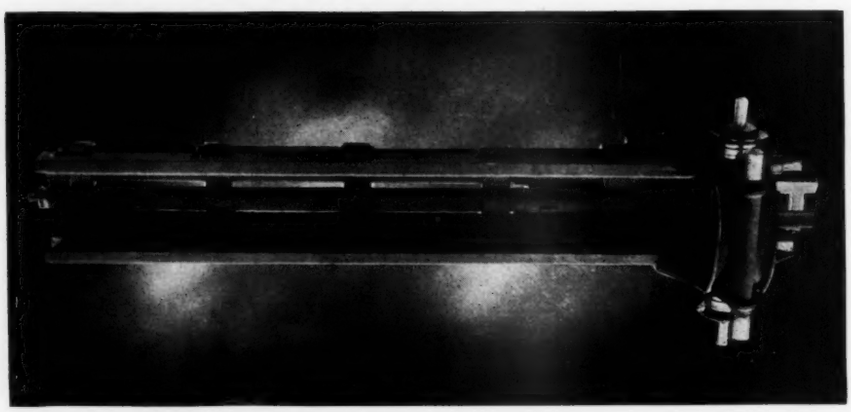


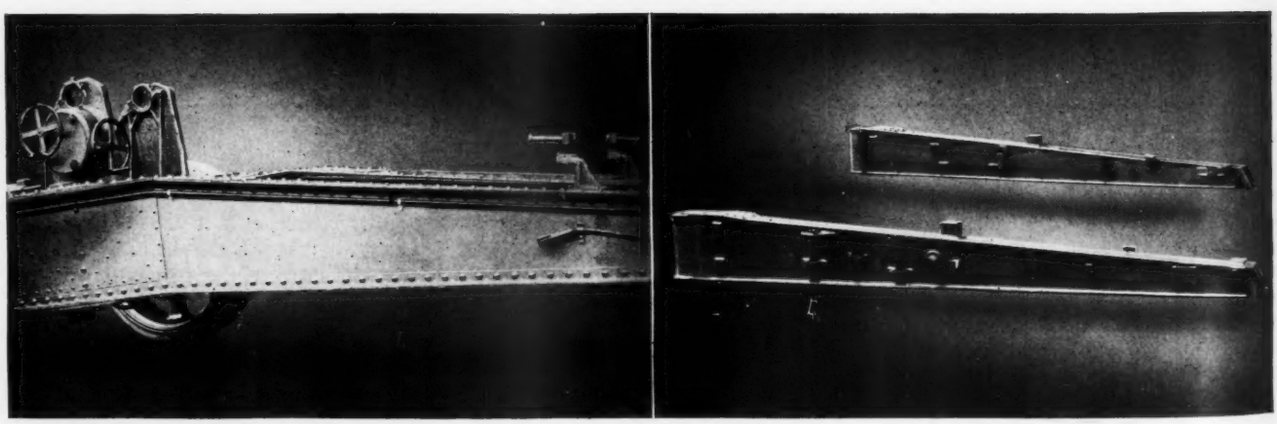
Two individually motor-driven, variable-delivery radial piston pumps, one for each hydraulic cylinder, are mounted on a common foundation at one end of the machine. Pushbuttons on the portable electric control panel start and stop the motors but the rate of speed at which the plungers move and the direction, up or down, is controlled by a pair of levers which can be placed at three different operating positions. These levers are connected to cables which actuate the pump shift lever extension thereby controlling the volume of oil delivered to the cylinders. Turnbuckles are provided to maintain tension in the cables. Since the speed of operation is controlled by varying the stroke and thereby the delivery of the pump, the control must be more sensitive than the lever control permits. For this reason the pumps are equipped with a hydraulic follow-up which operates as follows: as the pump control lever is shifted to one side or the other of the neutral position the control spindle is moved a proportional amount to one side or the other of its neutral position. This admits oil under pressure to one side or the other side of a piston which, in turn, moves a corresponding amount in the same direction as that of the control spindle, thereby causing the pump to vary its stroke in accordance with the movement initiated by the small control spindle. The pressure oil for the control circuit is furnished by a small gear pump driven directly off the main pump. Oil reservoir is fitted with fine mesh strainer. Four magnetic plugs give additional protection to pumps.

MODERN DESIGNS – Welded Parts For Guns

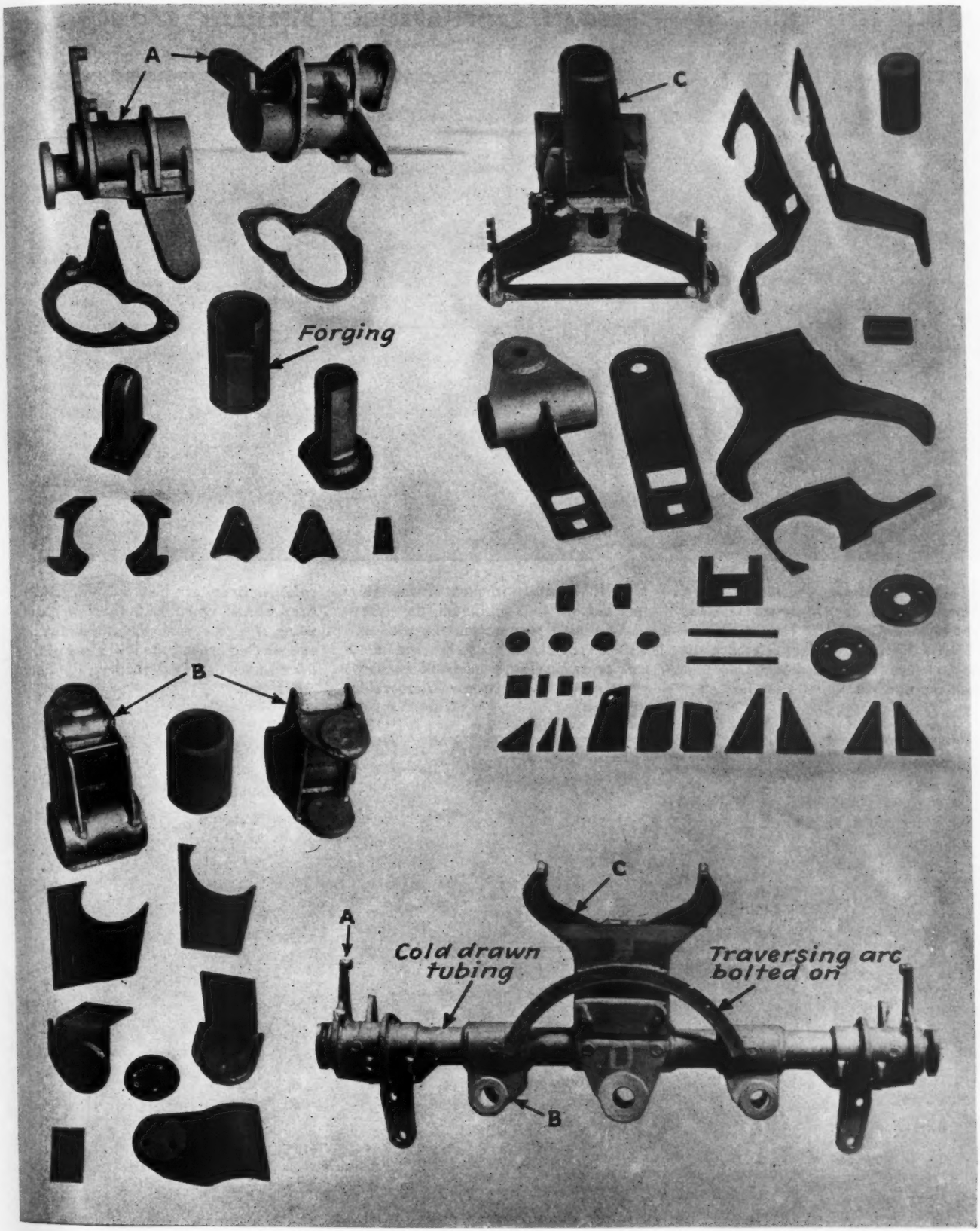
Cradle for gun recoil mechanism was formerly machined from a solid forged billet; the markings on the end of one of the forgings shows the metal which remained after machining. A portion of the finished cradle made of a forging is shown at the right. This cradle has now been redesigned and as manufactured at the Rock Island Arsenal consists of separate cylinders, yokes and guides assembled by welds and bolts as shown below.

Advantages of the new design are several. With the one piece cradle a defect which developed when the piece was near completion compelled rejection of the entire unit and consequent loss of all machining time up to that point. With the separate components rejections affect only the individual parts. This advantage also holds in repairing or reconditioning the recoil units. New parts can be installed where necessary without the need of scrapping the entire cradle. Another important consideration was that because of the size of the forging billet the forgings could be made only in the few shops that have large capacity machines. The present assembly can be manufactured in shops not equipped with special forging machines which is of particular advantage in time of emergency. Elevating arc is formed of two pieces of flat steel plate and one curved section welded together, gear teeth being cut after welding.



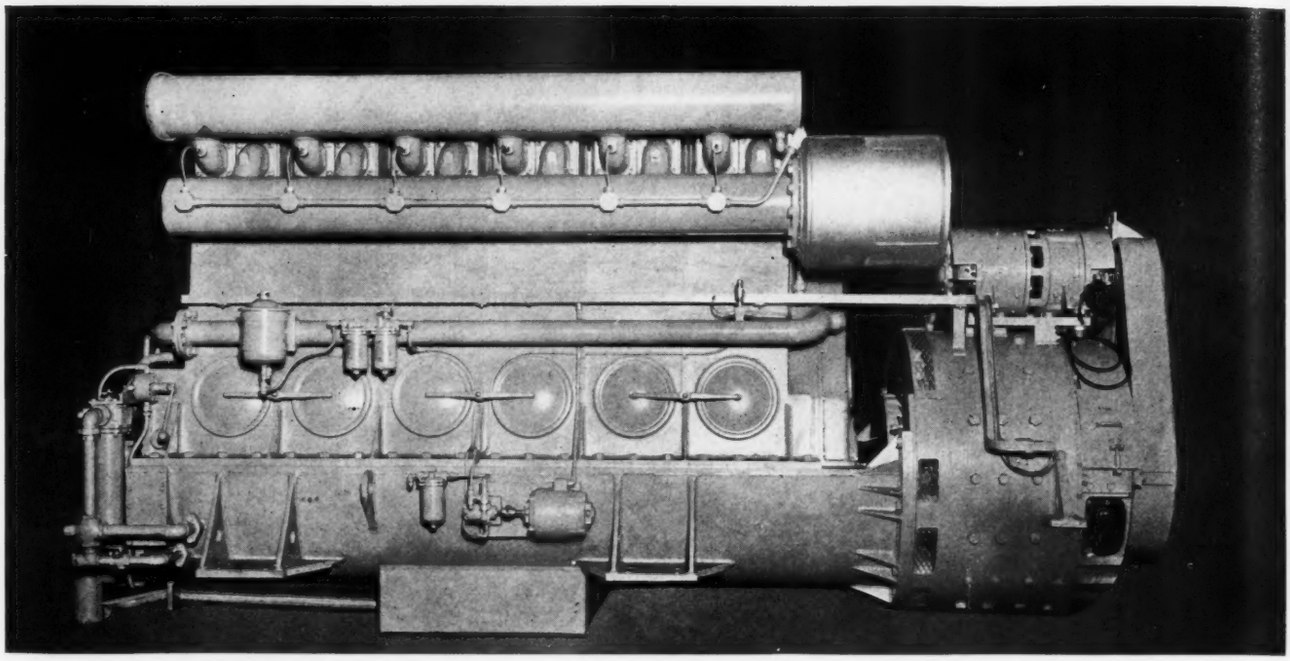


Trails of the 155 mm. gun carriage, formerly riveted girders, are now made of welded low alloy steel with consequent reduction in weight and cost while providing new features not available in the original design.

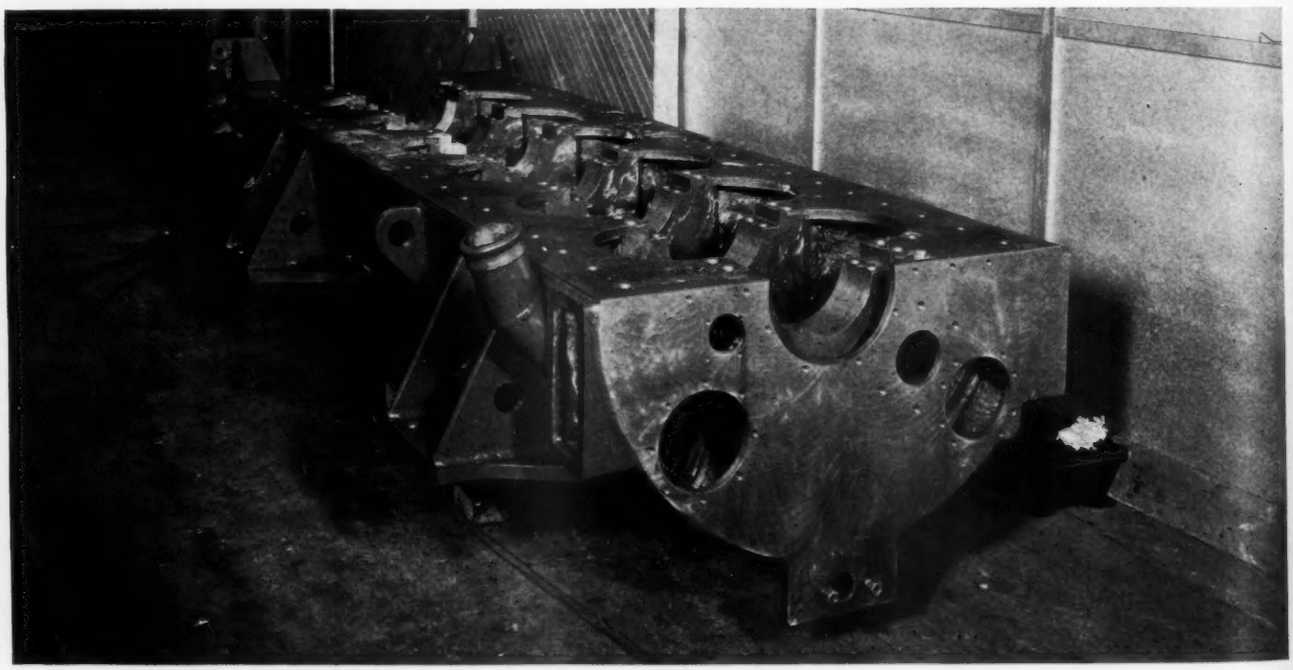


Formed sheet, bar stock, tubing and forgings are assembled by arc welding to make a gun carriage axle. Axles are stress relieved after assembly. Cold-drawn tubing used for the axle can be finished from the rough to desired dimensions in a centerless grinding machine, thus eliminating other machining operations. These axles are of a new type for the revised 75 mm. gun and perform multiple functions not required of the old axle. Part *A* supports the wheel spindles at the end of the axle; split trails hinge at *B;* top of firing jack is hinged on part *C*.

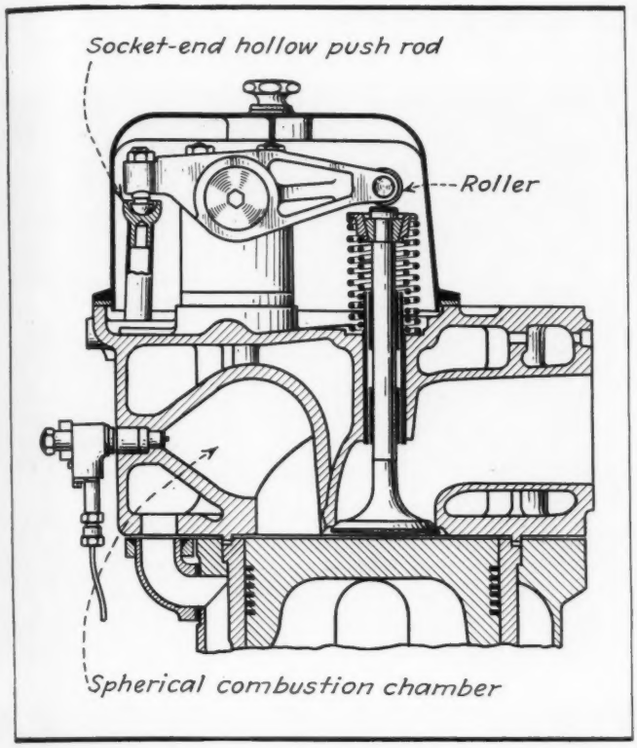
MODERN DESIGNS – Simplified Diesel Engine Design

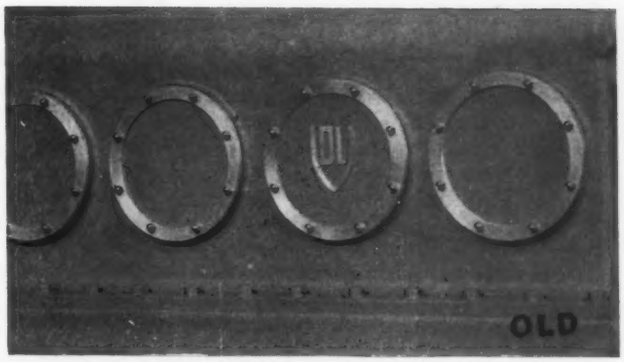


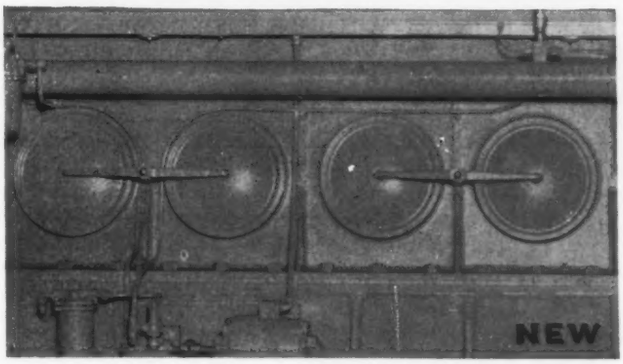
Special emphasis was placed on simplicity of design in the development of the new Model VO De La Vergne Diesels, with particular reference to maintenance and overhauling. To this end, the number of bolts for fastening parts that require frequent disassembly was reduced to a minimum, the necessity for high accuracy in the positioning of parts was largely eliminated and provisions were made to simplify all necessary adjustments. Removable cylinder liners are held by the cylinder head at the upper end, the lower end being free to expand longitudinally. A rubber ring around the lower end of the liner serves as a seal between the liner and frame.



Bedplate of the Model VO, is welded steel, semi-circular shape and reinforcements giving it exceptional rigidity. Heavy transverse webs support the main bearings, the caps of which are secured to the bedplate by internal through bolts. Cylinders are 12¾ inch bore, stroke is 15½ inches. All main bearings are adjustable by means of laminated shims. The bearing adjacent to the flywheel is longer than the other to take care of the greater load upon it, and is flanged to serve also as a thrust bearing to locate the longitudinal positioning of crankshaft.







Chain drive to the camshaft makes it unnecessary to maintain parallelism with the crankshaft to a high degree of accuracy, as would be required for a gear drive. Chain tension is adjusted by means of the idler mounted eccentrically. The camshaft bearing and follower guide block are accurately machined pedestals which are bolted to the frame. They are readily aligned with the cam and crankshaft but their positioning need not be highly accurate either longitudinally or laterally. The ball and socket end connections of the tubular valve push rods permit considerable freedom in the positioning of the camshaft pedestals. They are also doweled after alignment. This design also eliminates most of the highly accurate machining that would be required with conventional push rods and a gear drive to the crankshaft.

Older model De La Vergne Diesel had cast coverplates which required machining on both faces and drilling and a gasket cut for the studs. Each cover required 8 or 12 studs and nuts. In the new Model VO, the coverplates are stampings with curved lips that lock with the flued openings in the welded cylinder frame. Each pair of coverplates is held by an arm that is held at its center by one stud and nut. Only a simple ring gasket that fits in the curved lip of the coverplate is required. There is no machining on the coverplate.

The cylinder frame is entirely welded of flame cut plates and shapes.

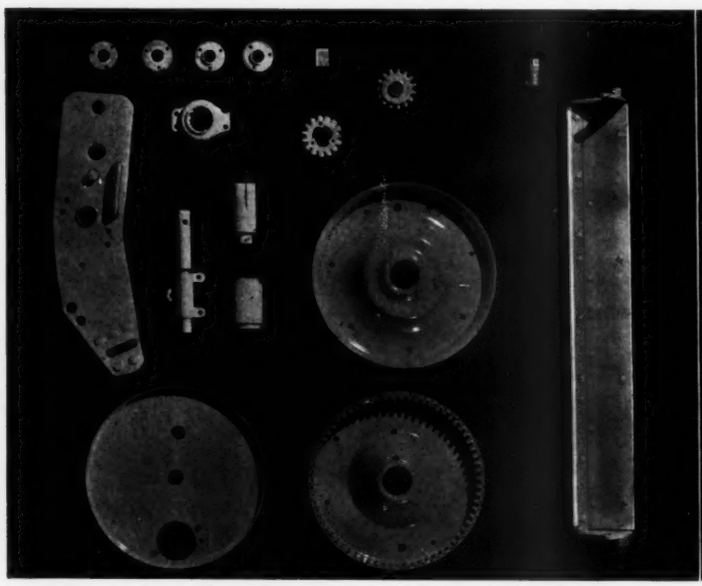
For engines up to 450 r.p.m. the pistons are annealed alloy cast iron, for higher speeds the pistons are aluminum alloy. Connecting rods are forged alloy steel, heat-treated. The crankpin bearings are steel backed babbitt, centrifugally cast and fitted with laminated shims to adjust clearances. The bearing at the wrist-pin end is a one-piece bronze bushing.

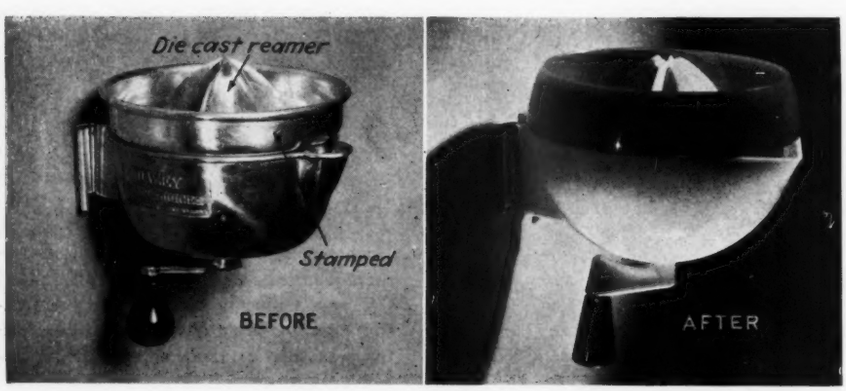
Pressure lubrication is furnished by a gear-type pump. The oil is drawn through a strainer and is delivered to a main lubricating oil header after passing through a filter and oil cooler. Branch pipes distribute the oil to the main bearings and then through drilled passages in the crankshaft and connecting rods to the wristpins. Cylinders are lubricated by oil thrown from the crankpin bearings.

MODERN DESIGNS – Pressed Steel Gears

Stampings replace castings in the new Jacobsen all-steel lawn mower for such parts as the wheel and its internal gear which drives the reel, pinions, bearing races and other parts. Wheels are rubber tired and molded rubber grips are used on the tubular steel handle. Cutter bar is hardened chrome nickel steel. Crucible steel bladed reel is balanced and mounted on adjustable ball bearings.

Internal teeth of pressed steel drive wheel are made from flat strip steel which is formed in ridges, cut to length equal to desired circumference and butt welded at joint. The internal gear is then case hardened after which it is peened into place inside the comparatively soft wheel stamping. This is a press operation. Pinions are made of eight identical stampings, case hardened, held together by two end stampings and three rivets. Two halves of wheel are riveted together in one punch press operation after the tire has been put in place, thus making for simple and rapid assembly. Split bronze bushings are rolled into place as the last operation on the wheel.





Formerly made of die-cast and stamped aluminum the Dazey juice extractor has been redesigned by George Moeller, chief engineer, and Barnes & Reinecke, to be molded in Bakelite polystyrene by Injection Molding Company. Metal parts which do not contact the juices are: insert for mounting on a universal wall bracket which is molded directly into the cup; a small V-shaped spring which holds the reamer on the axle, the rotating axle and a shaft for the handle. Plastics permit sales appeal through colors, and are odorless, tasteless and easy to clean.

BALL and ROLLER BEARINGS—II

Method of Selection

G. A. UNGAR
Consulting Engineer

WHEN selecting a ball, roller or needle bearing it is necessary to reduce all calculations to an equivalent standard for comparison. Manufacturers' catalog ratings are based upon different test methods and lengths of expected life—factors which lead to wide variations in catalog load capacity ratings for the same type and size of bearings of different makes. The calculation of theoretical bearing loads generally presents no problem and by analyzing shock and service conditions and selecting suitable multipliers as described in "Ball and Roller Bearings," P.E., Jan. 1941, pages 13–15 and 47-48, the required catalog rating can easily be computed.

However, the load calculation becomes more complex where the bearings are to operate at varying loads and speeds. Such conditions exist in machine tool applications, automotive transmissions and axles, roll stands, rotary pumps. For reasons of economy it is not always feasible to assume maximum speed and load as a basis for selection since this would lead to large and costly bearings.

As long as it is possible to make a reasonable assumption for the different loads and for the speeds at which these loads prevail, the following formula can be used to calculate an average effective bearing load P_{av} exerted at an assumed average speed N_{av}:

$$P_{av} = \left[r_1 \left(\frac{N_1}{N_{av}}\right) P_1^x + r_2 \left(\frac{N_2}{N_{av}}\right) P_2^x + \ldots r_n \left(\frac{N_n}{N_{av}}\right) P_n^x \right]^{\frac{1}{x}} \quad (1)$$

where

P_{av} = average effective load in lb.
r_1, r_2, etc. = fractional part of total bearing life during which loads P_1, P_2, etc. prevail
N_1, N_2, etc. = speeds corresponding to loads P_1, P_2, etc.
N_{av} = arbitrarily selected average operating speed in r.p.m.
x = exponent determined by the manufacturer, ranging between 3.0 and 3.33. A suitable average is 3.2.

By multiplying P_{av} thus calculated with the necessary safety or service factor, the required bearing rating at average speed N_{av} for the required bearing life will be found. It is now necessary to reduce the required rating to the same basis of hours expected life as is used by the manufacturer. This is done by the modified formula:

$$H = H_c \left(\frac{P_c}{P_{av}}\right)^x \quad (2)$$

where P_c is catalog load rating in lb., for an average life of H_c upon which manufacturer's load ratings are based; and H is the hours actual life. It is then only necessary to select from the manufacturers' load tables a bearing whose catalog rating is approximately the same as the required rating.

Selection Procedure

1. Compute average effective bearing load P_{av} from Formula (1).
2. Analyze service conditions; select type of bearing required. See P. E., Jan. 1941, p. 47–48.
3. Select shock load factor required under service conditions; multiply it by P_{av} as computed in step (1).
4. Compute load conversion factor $\sqrt[x]{H/H_c}$, from Formula (2). Multiply by load from step (3). Result is required catalog rating at speed N_{av}. Different factors must be applied when comparing bearings of different makes.
5. Select bearing size from catalog.

The following typical examples covering all types of modern ball and roller bearing applications show how this selection procedure can be applied:

Hydraulic Pump

A typical modern rotary gear pump is shown in Fig. 1. These pumps use double helical pump gears which eliminate axial gear thrust almost entirely, making it possible to dispense with bearings having thrust carrying capacity. It would be somewhat difficult to mount thrust bearings so delicately adjusted as to relieve the gear faces of their function as axial stabilizers. From a consideration of the operating conditions of hydraulic pumps, it will be seen that load carrying capacity, shock resistance and rigidity are of principal importance.

The pumps may be operated under hydraulic pressures up to 500 lb. per sq. in. producing radial loads which must be supported in a limited radial space. This limitation makes it necessary to dispense with separate inner races. Hardened shafts and lubricating oil used as hydraulic fluid make this possible.

While helical gears produce smooth mechanical meshing with smooth oil flow, the bearing loads change continually even at steady hydraulic pressure. There is also the possibility of hydraulic

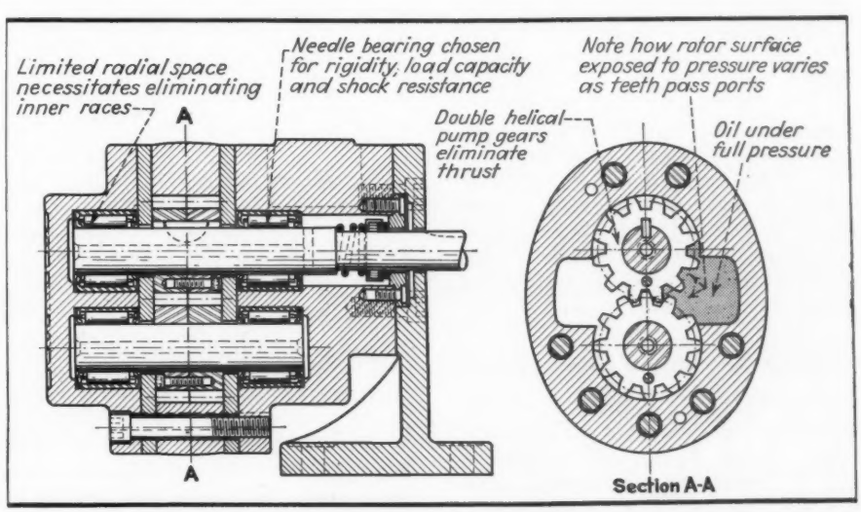

Limited radial space necessitates eliminating inner races
Needle bearing chosen for rigidity, load capacity and shock resistance
Note how rotor surface exposed to pressure varies as teeth pass ports
Double helical pump gears eliminate thrust
Oil under full pressure
A
A
Section A-A

Fig. 1—Needle bearings without inner races are chosen for this gear pump

shock from the control, from delayed relief valve opening or from air.

Bearings of maximum rigidity are required; heavy radial loads tend to deflect the shafts, and this deflection is less with long rollers than with short ones. Reduced shaft deflection means reduced rotor clearances and hence increased volumetric efficiencies. There are no problems of high speed. For pumps with ¾ in. dia. shafts the maximum speed is 1,750 r.p.m. and for larger shaft diameters 1,140 r.p.m.

All of these considerations point to needle bearings as the logical choice. For the selection of correct bearing sizes, make the arbitrary assumption that the pumps will operate at maximum pressure of 500 lb. per sq. in., at 250 lb. per sq. in. and at 50 lb. per sq. in. each during one third of the total life. Operating speed will be 1,750 r.p.m.

Each pump rotor has 1.6 in. face and 1¾ in. O.D. Its developed surface exposed to hydraulic pressure has a chord varying from 1 in. to 1¼ in. as the teeth pass the pump ports. At 500 lb. per sq. in. the total load on each of the two bearings therefore varies from 400 to 500 lb. or averages 450 lb. At 250 lb. per sq. in. and 50 lb. per sq. in. the bearing loads average 225 lb. and 45 lb. respectively. The average effective bearing load P_{av} from Equation (1) is calculated at constant speed (1,750 r.p.m.) as 325 lb.

Since the performance of a hydraulic pump depends upon the ability of the bearings to support the rotors without breakdown or wear, long life expectancy is imperative; 10,000 hr. of minimum life or about 3 years can be considered as satisfactory.

The required load rating of 10,000 hr. is found by multiplying the average bearing load of 325 lb. by 2.41, the load conversion factor, $\sqrt[x]{H/H_o}$ to change the minimum life expectancy from 10,000 hr. to 600 hr., upon which the needle bearing ratings are based, and by the shock factor of 1.5. This gives a required bearing catalog rating of 1,175 lb. at 1,750 r.p.m.

The needle bearing selected, with a catalog rating of 1,330 lb. at 1,750 r.p.m., is quite ample. At the normal operating speed of 1,140 r.p.m. the load rating of 1,465 lb. will provide a minimum life expectancy of about 22,000 hr. —6 to 7 years of safe operation.

Milling Machine Spindle Head

A good example of bearing selection for modern machine tools is the spindle head of a typical plain milling machine. Fig. 2 shows a section through armature shaft and spindle drive mechanism. The selection of bearing sizes has been based on a minimum life expectancy of 10,000 hr., which is the standard practice of some leading machine tool manufacturers.

SPINDLE MOTOR ARMATURE SHAFT—For the analysis of the factors underlying the selection of the two armature shaft bearings *A* and *B*, it is first necessary to consider the load. Assume that the motor is always operating at full 3 hp. load at a synchronous speed of 3,600 r.p.m. A load on pinion bearing *A* of 142 lb. and a load on rear bearing *B* of 45 lb. can be calculated.

Vibrations due to possible slight armature unbalance, from the pinion, from the mechanical brake and from the viscosity switch may cause some shock. The axial length of the bearings should be limited to reduce overhung loads and shaft deflection. The bearings should operate with a minimum of friction at the comparatively high shaft speed and they should compensate for slight misalignment caused in boring the housings.

For the pinion bearing *A* there is the additional requirement of axial shaft stabilization in both directions. Also, it should be shielded to prevent gear case lubricant from splashing into the bearing. The bearing choice is therefore a single-row ball bearing with deep groove raceways, double shields and an oil flinger to prevent oil from leaking into the armature.

For the rear bearing *B* at the brake end a single-row ball bearing is also the evident choice. There is no thrust and the bearing is mounted to permit axial float. A flinger prevents excess oil from fouling the armature.

Multiplying the calculated pinion bearing load with a shock safety factor of 2, a bearing rating of 285 lb. required for 10,000 hr. of minimum life expectancy is obtained. The bearing selected from the manufacturers catalog has a rating of 655 lb. at 3,600 r.p.m. for a 1,000 hr. minimum life expectancy, and from Equation (2) a rating of 325 lb. at 3,600 r.p.m. for a 10,000 hr. minimum life expectancy.

The choice of the rear bearing *B* is a medium series. While a considerably smaller bearing would have served as well, there is a decided advantage in specifying both bearings with the same shaft and housing dimensions. Furthermore the medium series of bearings selected is less subject to damage in mounting.

The considerations for the selection of the small bearing *C* inside the me-

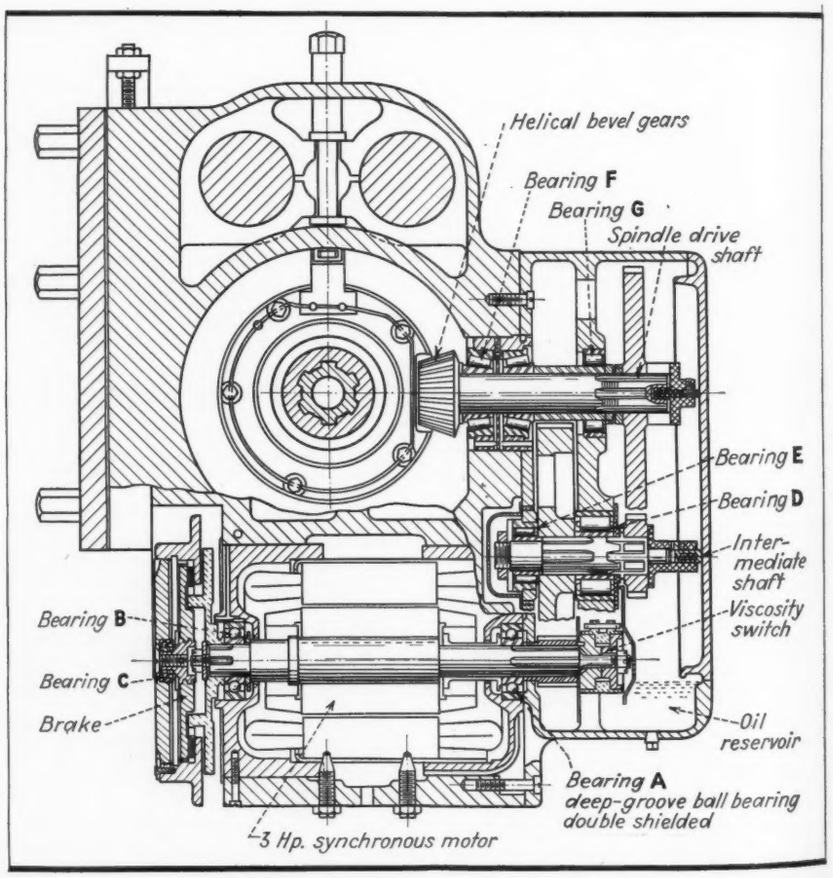


Fig. 2—Varying speeds and loads are present in milling machine spindle drives

chanical brake shown at left are primarily low starting friction, a certain thrust capacity and self-contained design. The choice is therefore a radial single-row ball bearing, shielded to retain lubricant.

INTERMEDIATE SHAFT—An investigation of the loads imposed on the bearings discloses a considerable range at the same speed of 642 r.p.m. depending on the ratio of the change speed gears used. This load range is particularly evident at the outboard bearing *D* near the change gear, where there is an increase due to the overhanging gear and close spacing of bearings.

The shock effect due to spur teeth and the relatively loose spline fit of change speed gears must be taken into account when making the selection of suitable types and ratings.

These factors, as well as shaft rigidity, point to the selection of cylindrical or tapered roller bearings. Since axial loads are negligible and stabilization is accomplished satisfactorily by plain washers, cylindrical roller bearings are the logical choice.

The bearing load calculations, if based upon the assumption that maximum loads prevail during the entire expected bearing life, would lead to excessive bearing sizes. It is therefore advisable to make assumptions for the prevalence of bearing loads.

In order to calculate the substitute bearing load P_{av} from Equation (1), assume that during the desired life of 10,000 hr., 2,500 hr. will be at maximum load, 2,500 hr. at minimum load, and 5,000 hr. at an average load (produced when intermediate and spindle drive shafts operate at 642 r.p.m.).

With a normal 3 hp. motor load there is a constant torque of 295 in.-lb. at the constant shaft speed of 642 r.p.m. At the minimum spindle speed the calculated load on outboard bearing *D* is 600 lb., at the maximum spindle speed 150 lb. and at average spindle speed 225 lb. The corresponding loads at the inboard bearing *E* are 167 lb., 1 lb. and 30 lb., respectively.

For bearing *D* we therefore obtain from Equation (1) $P_{av} = 372$ lb. The rating of the bearing selected from the catalog for this location at 642 r.p.m. is approximately 1,000 lb. for an average life expectancy of 5,000 hr. or about 1,200 hr. minimum life expectancy. From Equation (2) we obtain for 10,000 hr. minimum life expectancy a load rating of 530 lb. The shock safety factor of $530/372 = 1.43$ is satisfactory.

Selection of the inboard bearing *E* is based on a similar procedure, and P_{av} is calculated as 101 lb. The rating of the selected cylindrical roller bearing at 642 r.p.m. is approximately 500 lb. for 1,200 hr. minimum bearing life expectancy or about 265 lb. for 10,000 hr. minimum life expectancy. The shock safety factor of $265/101 = 2.62$ is ample and would justify a smaller bearing, but smaller shaft size would reduce rigidity.

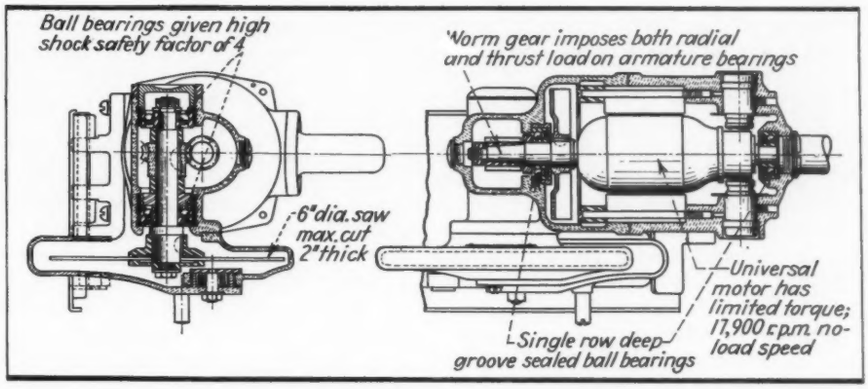

Ball bearings given high shock safety factor of 4
Worm gear imposes both radial and thrust load on armature bearings
6" dia. saw max. cut 2" thick
Universal motor has limited torque; 17,900 r.p.m. no-load speed
Single row deep-groove sealed ball bearings

Fig. 3—Bearings in portable saw have minimum friction over wide speed range

SPINDLE DRIVE SHAFT—The analysis of loads and speeds discloses a wide range of both. A tabulation of the loads and speeds is given below. The loads for the pinion bearing *F* are equivalent to radial loads for combined radial and thrust loads, the latter resultant from the helical bevel gear.

SPEED R.P.M.	LOAD *F*	LOAD *G*	*r* FROM EQUATION (1)
123	3,900 lb.	1,150 lb.	0.25
3,267	158 lb.	111 lb.	0.25
642	1,027 lb.	255 lb.	0.50

Due to pinion overhang the radial load on the bearing *F* is increased and the tendency for shaft deflection is increased. The overhung change speed gear also causes increased radial loads on outboard bearing *G*. The shock effect from the two overhanging gears is pronounced. All these considerations point to the choice of rigid roller bearings.

For bearing *F* we find from Equation (1) the substitute load P_{av} at the assumed average speed of 642 r.p.m. as follows:

$$P_{av} = \sqrt[3.2]{0.25\left(\frac{123}{642}\,3{,}900^{3.2} + \frac{3{,}267}{642}\,158^{3.2}\right) + 0.5\,\frac{642}{642}\,1{,}027^{3.2}} = 1{,}660 \text{ lb.}$$

For the pinion bearing *F* in which thrust in both directions due to spindle reversals must be supported, two tapered roller bearings are chosen from the catalog. The two bearings selected have a combined radial load rating of 3,320 lb. at 642 r.p.m. with a minimum life expectancy of 3,000 hr. or about 2,220 lb. rating at 10,000 hr. minimum life expectancy. The shock safety factor is 1.34.

Similarly for bearing *G* we find P_{av} at 642 r.p.m. to be 422 lb. For this bearing the cylindrical roller bearing with extended inner race is selected from the catalog. The load rating of this bearing is 1,625 lb. at 642 r.p.m. for about 1,200 hr. minimum life expectancy, or calculated from Equation (1) about 850 lb. for 10,000 hr. minimum life expectancy. The shock safety factor of 2.02 is ample and a smaller bearing might be chosen if a smaller shaft diameter were possible.

Portable Electric Saw

In Fig. 3 is shown the Stanley Electric Tool Company W-60 portable wood saw, with a 6-in. blade and a capacity of cutting up to 2 in. lumber. For the selection of suitable bearings the following assumptions can be made. The loads imposed will be based on full motor torque. With an average bearing life of 3,500 hr. the bearings will probably be good for a service life of over 3 years, since portable saws are not used for continuous service nor for maximum lumber thickness.

For the selection of suitable types the principal requirement is minimum friction over a wide range of speeds. This is important because the torque output of universal motors is limited. There also is the requirement of slight self-alignment and thrust carrying capacity. Also, built-in felt seals for the armature bearings and for the bearing near the saw blade are necessary. These requirements point to a single-row deep groove radial ball bearings as a logical choice.

The motor at full load develops 1 ft.-lb. torque at 2,620 r.p.m. The no-load speed of the motor is 17,900 r.p.m., hence the requirement for high speed performance of the bearing. The worm gear has a 5.33:1 ratio and a helix angle of 15 deg. 21 min. The worm bearing of the armature shaft, which carries both radial load and worm thrust, is subjected to 135 lb. calculated equiva-

lent radial load. With a shock factor of 1.5 the required load rating is 203 lb. at 2,620 r.p.m. The bearing selected from the manufacturer's catalog has a rating at this speed of 210 lb.

The rear bearing of the armature shaft, a ball bearing, has only the function of a heavy bearing subjected to about 15 lb. radial load. The rating of this bearing being also 210 lb. at 2,620 r.p.m., a smaller bearing could be used if a reduction of the shaft diameter were desirable.

The wormwheel thrust on the saw arbor is carried on the inboard bearing while saw stabilization in the other direction is provided in the outboard bearing. The radial loads on the saw arbor are calculated with the assumption of a maximum saw resistance in the direction of the cut of 40 lb. corresponding to the limit of manual feed pressure.

Because of shocks due to the action of the saw, possible nails in the wood, and saw unbalance, it is necessary to use the exceptionally high safety factor of 4 in computing the load ratings. The calculated equivalent radial load of the inboard bearing is 134 lb. The required load rating is $4 \times 134 = 536$ lb. The rating of the bearing chosen is 550 lb. at 490 r.p.m. and 3,500 hr. average life.

The calculated radial load of the outboard bearing is 103 lb. at the full load speed of 490 r.p.m. With a safety factor of 4 the required load rating becomes 412 lb. and the selected bearing, with 410 lb. capacity at 490 r.p.m. and 3,500 hr. average life expectancy is satisfactory.

Sheet Metal Cutting Shear

The bearing installation chosen here is for the lower shear head of a Stanley Electric Tool Company Model B Unishear. This machine is used for shearing sheet steel (up to ¼ in. boiler plate) along straight or curved lines.

The bearings shown in Fig. 4 support an eccentric shaft operating at 1,200 r.p.m. which in turn carries an oscillating rocker for the actuation of the shear blade.

In addition to the shearing resistance imposing considerable bearing loads, there are the inertia loads from the reciprocating rocker and connecting rod parts. On the eccentric bearings proper there is the additional effect of centrifugal and oscillating forces on cages and rollers. These requirements clearly point to the choice of tapered roller bearings in pairs, adjustable to eliminate all play.

Shears of this type are rarely in continuous service. Also the bearing loads vary greatly because of the variety of sheet metal gages handled and because of the change in cutting speeds. By using maximum estimated bearing loads and choosing bearings with a minimum life expectancy of 3,000 hr., service will be from 3 to 5 years.

The actual load on the rocker when cutting soft ¼-in. steel at the maximum feed of 15 ft. per min. is approximately 2,500 lb.; for $\frac{3}{16}$ in. steel about 1,750 lb.; and for ⅛-in. steel about 1,250 lb. To this must be added about 250 lb. due to inertia forces resulting from unbalanced high speed reciprocation.

To determine an equitable substitute bearing load P_{av}, assume that the shear will handle ¼-in., $\frac{3}{16}$-in. and ⅛-in. sheet steel, each thickness during one third of its total operating life. The load per bearing, P_{av} is then calculated as 995 lb.

In view of the oscillating character of the load and the possibility of considerable shock, an ample shock factor must be used. For the two outside bearings the shock factor required is 2.5 while for the eccentric bearings the safety factor used is 3.5 because of the added highly destructive effects of the gyration of rollers and retainers.

The necessary rating of each outside bearing is therefore $2.5 \times 995 = 2{,}475$ lb. which is properly taken care of by the selected standard bearings, which have 2,885 lb. radial rating at 1,250 r.p.m. for 3,000 hr. minimum life expectancy.

For each eccentric bearing the required rating is $3.5 \times 995 = 3{,}475$ lb. The bearings selected from the catalog for this location have a radial rating of 3,680 lb. at 1,250 r.p.m. with 3,000 hr. minimum life.

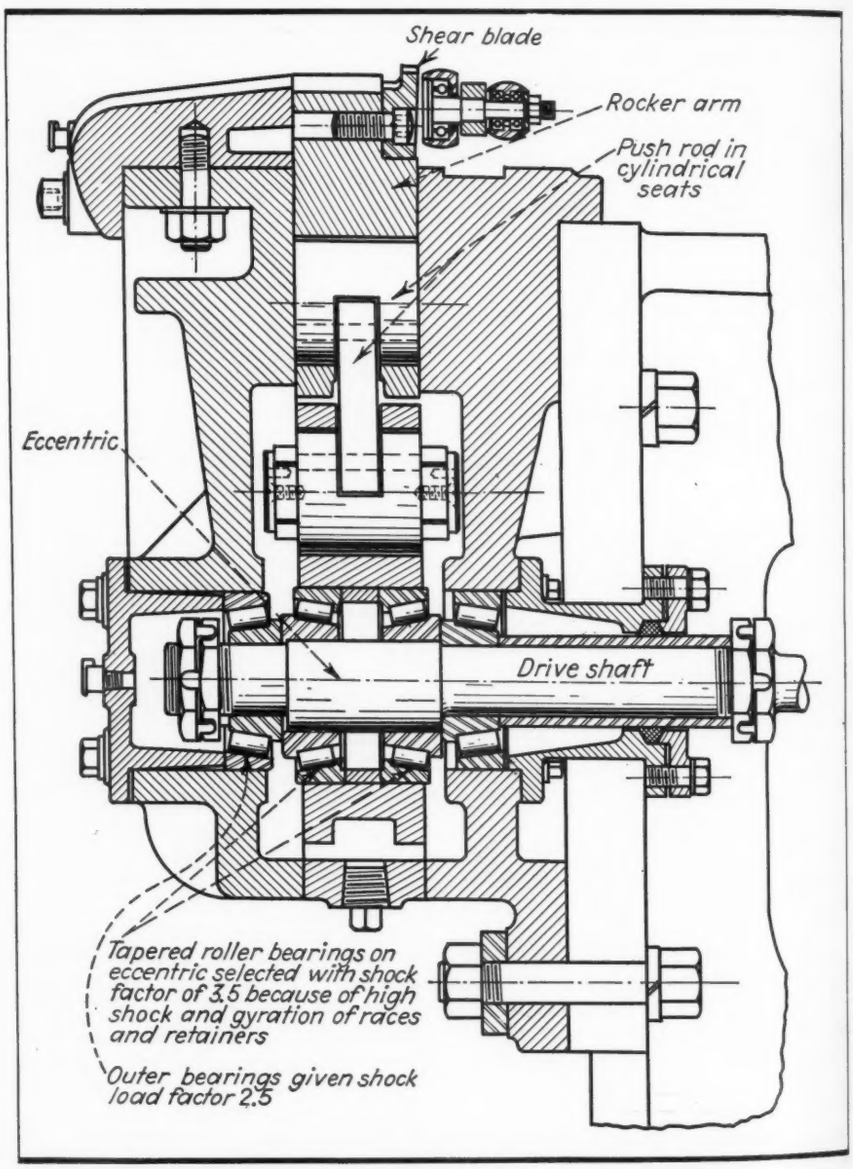
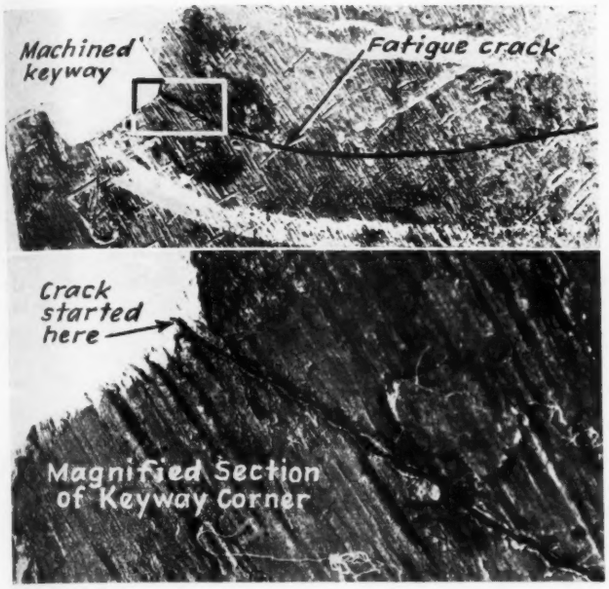
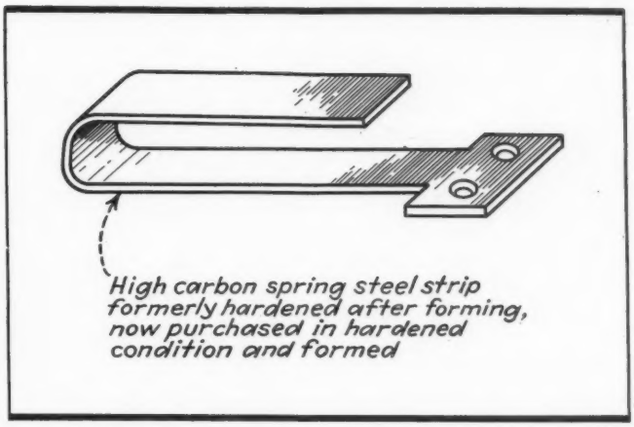

Shear blade
Rocker arm
Push rod in cylindrical seats
Eccentric
Drive shaft
Tapered roller bearings on eccentric selected with shock factor of 3.5 because of high shock and gyration of races and retainers
Outer bearings given shock load factor 2.5

Fig. 4—Tapered roller bearings in Unishear are chosen for high shocks loads

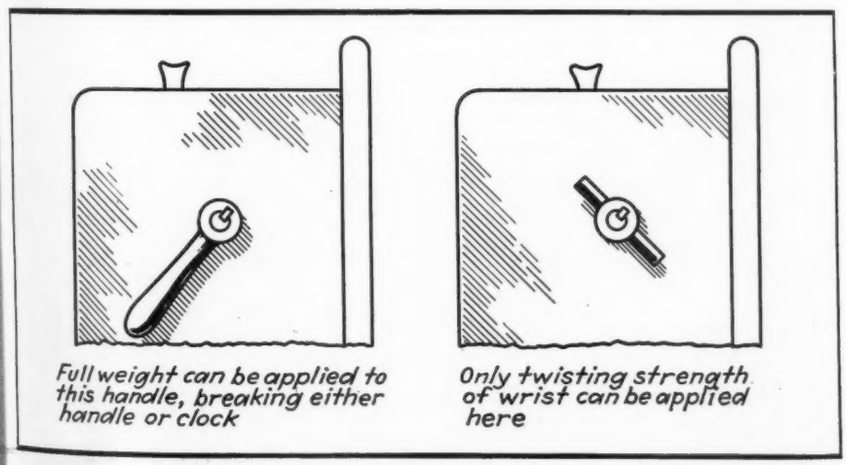
Causes and Cures



Fatigue crack which started in the sharp corner of the machined keyway in a forged part. Crack started ½ in. from the face of the part and followed along the corner for the greater portion of the length of the keyway. Keyways are likely places for such fatigue cracks to start and failures can be avoided only by specifying smooth circular fillets in the corners. Corners of the key should be rounded to a larger radius than the keyway fillet in order to avoid work-hardening of surface metal in the corner of the keyway.



Flat spring of the type shown, used in a piece of low-priced office equipment, showed discoloration indicating uneven heat-treatment. Procedure was to purchase annealed high-carbon strip steel, punch and form the spring, heat-treat to desired Brinell hardness, assemble. But, after making small changes in the die to allow for additional "springiness" of the material, it was possible to purchase the strip with the desired hardness, making it necessary only to form and assemble. Not only was production cost reduced but uneven heat-treatment was eliminated and the spring had a good appearance. Although not generally realized, there are many cases where high-carbon steels can be purchased with a suitable hardness and "springiness" for a design and yet can be worked without annealing or subsequent heat-treatment.


Full weight can be applied to this handle, breaking either handle or clock
Only twisting strength of wrist can be applied here

Workmen "punching" a well-known model of time clock found it too easy to break off the cast operating handle, either by exerting their whole weight or by kicking at it. Substitution of wrought iron handles only aggravated the situation, since some expensive part of the clock mechanism would break if the handle withstood the abuse. This minor form of sabotage was finally eliminated by substituting the type of handle shown. Force could only be applied by twisting, and wrists are not strong enough to do any damage.

Crankshafts for well-drilling machinery were formerly made of carbon steel. Because of the severe fatigue loads placed on the crankshaft in drilling service, breakages of this part were frequent. Cure was adoption of S.A.E. 3140 nickel-chromium steel.

The embrittlement effect of molten metals is often forgotten in design. For example, a label soldered on aircraft tubing caused failures in the section near the soldered joint. Elimination of the label put an end to failures. The same phenomena applies where other molten metals such as zinc, lead, or tin come into contact with stressed members. Similarly two galvanized sheets riveted together may show cracks around the joints.

PRODUCT ENGINEERING will pay a minimum of $3 for each example published in Causes and Cures. Where illustrations are necessary, include drawings, rough sketches or photographs.

SPRAYED FABRIC FINISHES

For Decorating, Insulating and Sound-Deadening

FLOCKING, or spraying of short fabric fibers on an adhesive binder, a method for producing a velour or suede-like finish on any base material, has been in use for years in applications such as protective coatings for the under surface of book-ends, lamps, telephones and ornaments, and for finishing the inside of cases, instrument housings and the like. However, development of new automatic and semi-automatic flock spraying equipment has stimulated the use of this finishing method in the automotive, aircraft and metal-working fields not only for its decorative qualities but also for its value as sound and heat insulation.

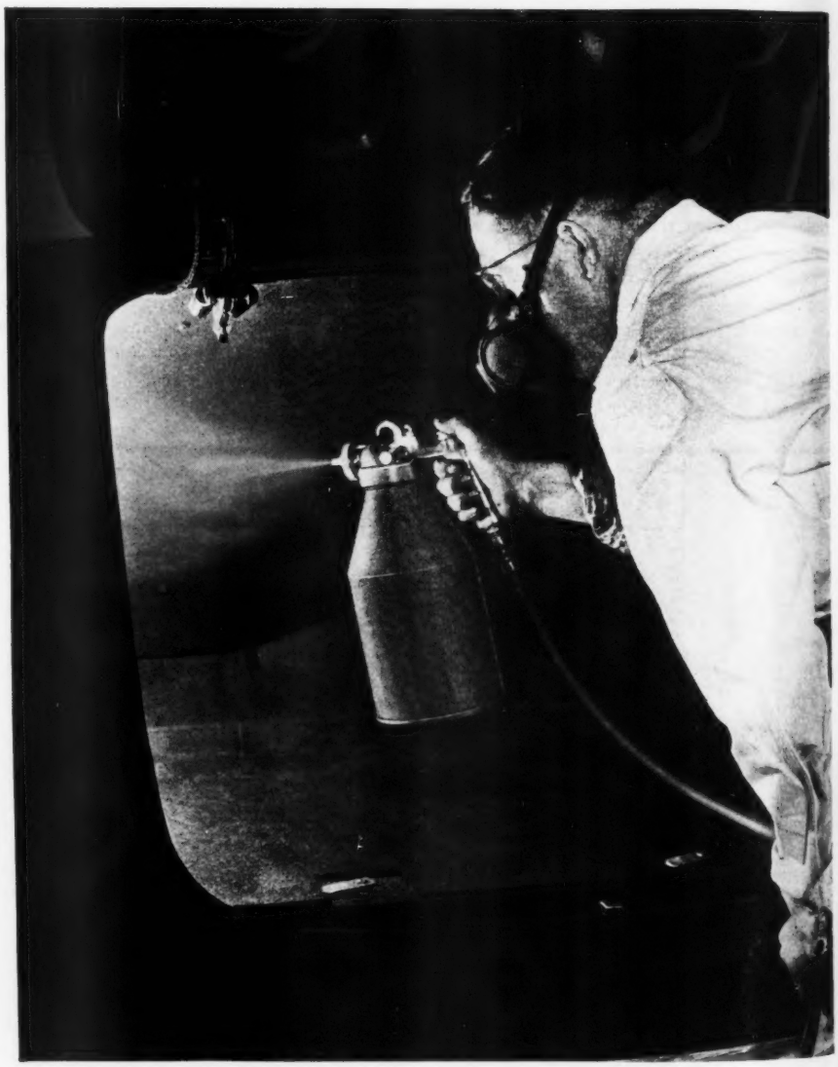
With the new fully automatic machinery, all that is required is that the operator place the part on a conveyor which carries it into booths where first the adhesive binder and then the flock is applied. In a phonograph disk application, the disk is manually placed on a conveyor spindle and carried into the adhesive spray booth where it is rotated under the spray nozzles turned on automatically by an electric limit switch. The conveyor then carries the disk into the second booth where flock is applied by nozzles turned on automatically by an air valve. Here again the disk revolves while passing under the nozzle. Air jets remove excess flock, which is rinsed out and recirculated. Average rate of operation of this machine is ten 12-in. disks per min., or 4,800 per 8-hr. day. Similar equipment is recommended where areas to be flocked are relatively small in size, a single color is used, and production lots are large.

For small objects where production lots are small or where flocking colors must be changed occasionally, a manually operated airbrush can be used for application of the binder and a hand air-gun for applying the flock. Where large areas are to be sprayed, the flock is sometimes applied by large nozzles at comparatively low pressures with large volumes of air. For inaccessible places or for large production work, as the spraying of glove compartments and the trunks of automobiles, a mechanical feed hopper and special gun is used for applying the flock. Hopper and gun are connected by an extensible length of hose so that the operator can move around the work.

Special decorative effects are obtained with stencils through which the binder is sprayed on the base in the same manner as in other paint-work. The flock subsequently applied adheres only to the tacky binder. For accurate work or complicated designs a silk screen method, similar to that used in enamelling and other decorative work, is adaptable to applying the binder, after which the flock is applied with the conventional gun.

Flocking has recently been adopted as a low cost method for lining glove compartments and trunks of automobiles, as a decorative finish for containers, stoves, radio screens; as a heat insulator in motion picture projectors. Airplane manufacturers have been experimenting with flocking for interior design. As such, it has been used in decorating seats of Douglas 21-passenger airplanes built for the Braniff airlines. In this field it serves not only a decorative purpose but it also relieves the "cold" feeling of metal, particularly when the plane is at higher altitudes. Other aircraft manufacturers are now experimenting with the application of flocking for decoration and as a means of furnishing desired acoustic properties.

In the automotive field, automotive



Applying flock to Studebaker trunk compartment interior

Velour effect on the back of the seats in this Douglas plane is obtained by applying flock to the sheet metal panels

engineers have been flocking trunk and glove compartments to obtain a low cost yet good-looking durable finish. Other possibilities may lie in applications where ordinary fabric is difficult to install and where the use of flock might make for improvement or lower production cost; one possibility under consideration is the use of rubber floor mats sprayed with flock thus combining the resiliency of one material with the appearance of another.

Flock is now furnished in accurately controlled cut lengths rather than in the ground form previously used which, because of its variable particle size, was difficult to apply without giving a "matted" effect. The flock is produced in cut lengths so that it can be driven like an arrow into the binder to give good adhesion with a fluffy, even-appearing surface. General improvement has been made in synthetic binders and formulas developed which make it possible to apply flock to rubber. There are now available to the designer a wide variety of materials, colors, and binders which can be applied in several combinations. Materials are furnished in stock lengths of 1/32, 1/16, 1/8, 3/16 and 1/4 in. in rayon, wool and cotton, variously colored. For special purposes such as heat insulation, asbestos, animal hair or fiber may be used. Glow beads may be used for decorative purposes or as visible markers in darkness.

In selecting material, consideration must be given to the service required. In general, flock materials will react to atmospheric and other conditions in the same manner as the basic cloth of which the flock is made. Experimentation will probably be necessary to determine which of the five available lengths of flock give the desired appearance and thickness of finish. The length of the flock affects not only appearance but also service life, the latter increasing with the length of the flock and with a heavier pile. The 1/16 and 1/8 in. lengths are best for all-around work where finish alone is concerned. If acoustic properties are desired it may be necessary to apply longer flock or more than one coat of flock to the base material. Four, five and six layers have been successfully applied.

For fine work and close registration of color 1/32 in. flock is generally specified. For large areas, 1/16 in. length flock is used because it gives a fairly long nap and is easy to apply.

Rayon has the longest nap of the materials but is not very opaque. To add opacity and to reduce cost, as much as 25 percent wool or cotton of the same color may be added. Wool is more opaque than rayon but has a shorter nap. Cotton covers well but has practically no nap at all. In those cases where the nap must stand up and give a good velour-like appearance, such as in radio screens, the base material is vibrated while the flock is being applied. This vibration causes the nap to stand upright thus giving the screens an appearance of solid plush. Rayon flock will cover about 100 sq. ft. per lb. on the first coat and about 120 sq. ft. on the second coat. Wool and cotton flock will cover about 90 sq. ft. on the first coat.

Rayon, wool and cotton flock are available in bright red, orange mineral, medium yellow, emerald green, medium blue, gold, silver, black and white. Wool and cotton are also available in orange and green while the rayons are also available in nine pastel shades. To secure other shades of color the flock is mixed like paint. The flock materials mix as well as ordinary oil colors and can be varied to different shades by proper proportioning of the basic colors.

Heavy synthetic oil-varnish binders formulated with a retarder to slow up drying are sometimes used to hold the flock to wood, metal, rubber, cloth, paper or other bases. These materials are known in the trade as Vellunite, Kem, Izarine, F. Weber Flock Binder and Flextite. Latex has been used successfully to bind flock to a rubber base. A primer is generally needed before spraying the binder on metal. On porous surfaces a coat of filler and sealer must be used before applying the flock binder.

Color of the binder is matched to the color of the flock unless unusual effects are desired. To enhance the finish a shading of a darker color may be applied around the openings and edges. Where it is desired to get a decorative effect by super-imposing one color on another a clear binder is sprayed on the first layer of flock and second layer of

flock applied immediately before the clear binder can be absorbed.

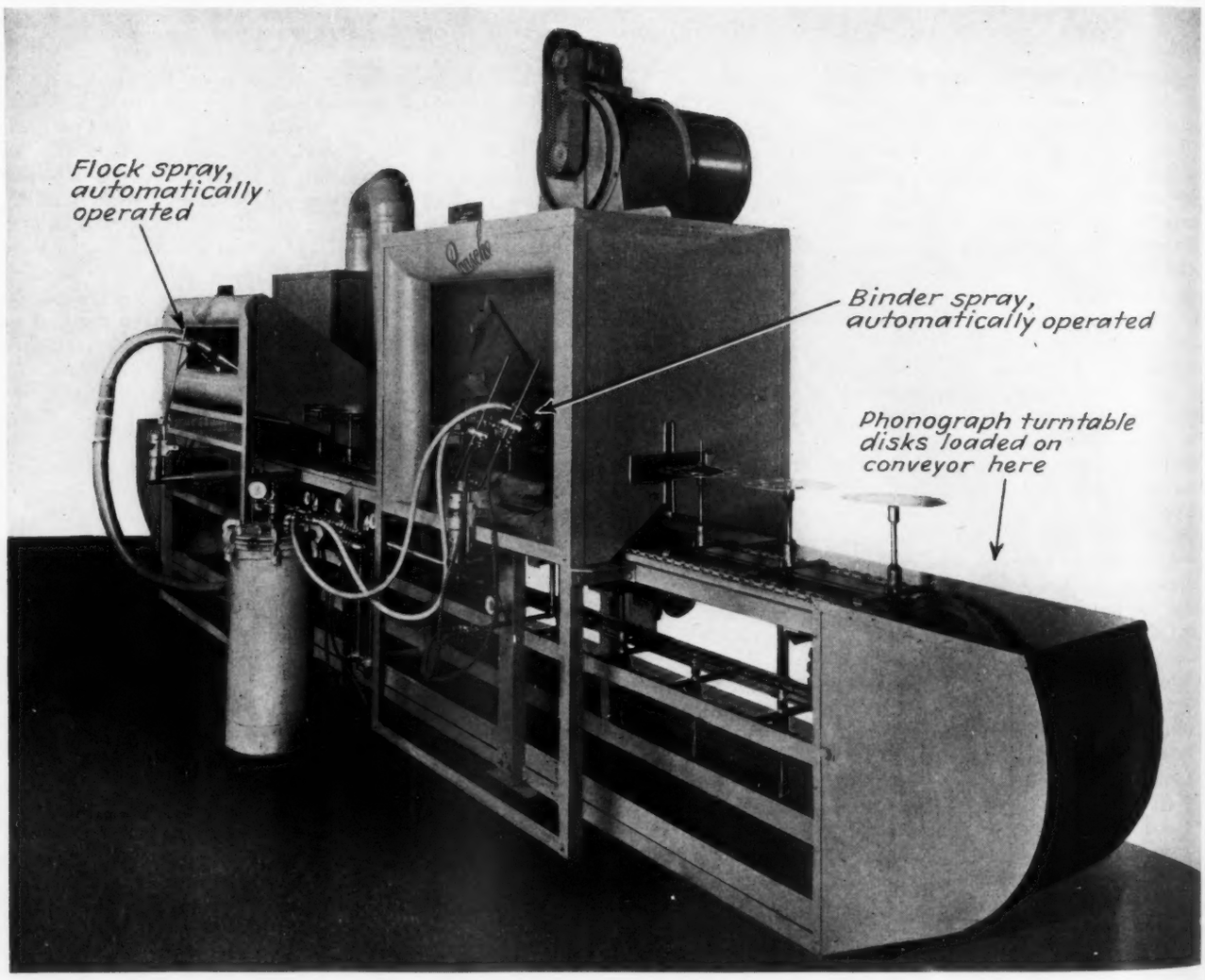
Application of these materials is similar to paint. The binder will cover 250 to 300 sq. ft. per gal. As the area is increased the binder should be mixed with more retarder to give slower drying.

Base materials such as metals, wood, paper, fabrics, cork, rubber and others have successfully been flocked. Flock serves as a protective and decorative finish for metals and has the additional feature of eliminating the "cold" feeling which they possess. Experiments are now underway on applying a sprayed cloth covering directly to Airfoam rubber cushions. Other experiments are being made to give rubber floor mats a plush or velour-like appearance by spraying with flock of the desired color. One company is "printing" the flock on paper for decorative purposes at a high production rate.

Resistance to wear is dependent primarily on the qualities of the basic materials from which the flock is made, the thickness of the pile and sometimes on the quality of binder. In a particular automotive application it was found after experiment that rayon gave the most satisfactory results as far as long-life wearing qualities are concerned. Wear properties can be improved by using a longer or heavier flock or a heavier pile. Experiments on the application of flock directly to Airfoam cushions, mentioned above, indicated that the type of binder was most important from a wear standpoint. In this case it was necessary to find something which would vulcanize to the rubber. Latex and Flextite were satisfactory. It was also found desirable to change the shape of the cushion so as to eliminate square edges which take heavy wear. Even rubber base binders would not stand up under the mechanical effects of wear on such corners.

Flocking has been used to enhance the appearance of low cost woods. The operation in this case is to first prepare the surface so that it is dry and smooth, after which a white lacquer undercoat reduced about 40 percent with a good grade thinner is used as a filler. When dry, this coating is sanded lightly and the binder is then applied. One successful application uses a synthetic silk screen mixture using equal parts of clear synthetic varnish reduced to spraying consistency with a slow drying synthetic reducer. Flock is then applied with low pressure, about 30 to 40 lb. and a large volume, 6-7 cu. ft. per sec., of air.

Whether applied on metal or wood the binders require up to 24 hr. to dry unless an oven is used. When heat is used, temperature must be controlled so as not to harm either the flock or base material. On metal, drying time may vary from 45 min. at 250 deg. F. (which will not harm rayon) to 2 hr. at 165 deg. F. Wood, however, has a tendency to crack under heat and to cause the resins or liquids to come to the surface. As a result, the finish blisters and flakes off so that a maximum of 150 deg. F. drying temperature is advised for flocked wood parts.


Flock spray, automatically operated
Binder spray, automatically operated
Phonograph turntable disks loaded on conveyor here

Fully automatic machine which applies black or brown rayon coating to phonograph turntables at rate of 4,800 per 8-hr working day

DRAG-LINK MECHANISM—II

Effect of Link Lengths on Angular Velocity

ALBERT SHIELDS and ERNEST ROSMARIN
S & S Corrugated Paper Machinery Company, Inc.

EFFECTS upon the velocity diagram of drag-link mechanisms, obtained by altering the proportions of the members comprising the linkage for two cases were analyzed in the first installment of this article, which was presented in the March number of PRODUCT ENGINEERING. Case I discussed the effects obtained by altering the length of the link and also of the eccentricity, when the follower crank is equal in length to the driving crank. Case II discussed the effects obtained by altering the length of the follower crank and the eccentricity, when the link is equal in length to the driving crank.

This part of the article will present an analysis of the velocity effects for Case III in which the follower crank equals the length of the link but of a length different from that of the driving crank; and for Case IV in which the follower crank, the link and the driving crank are all of the same length.

Case III is fundamentally the same as Case II, a change from driven to driving crank in either case making it similar to the other. However, to find R_3 from R_2, it would not only be necessary to multiply R_2 by p^2, but θ must be replaced by γ. Since this last step would require a derivation much like the one already made, this combination will be treated as a distinct case. By setting $m = p$ in Equation (10) the ratio for equal link and follower crank becomes:

$$\text{Ratio}_3 = \frac{1 - n\cos\theta}{K} + \frac{n\sin\theta}{\sqrt{4p^2K - K^2}} \quad (13)$$

In this equation, p appears only in the expression under the radical sign. As K or $(n^2 + 1 - 2n\cos\theta)$ is always a finite quantity, the second term of the equation will approach 0 as p approaches infinity, and R_3 will approach the ratio of the crank and slotted arm mechanism. In other words, as p increases, the characteristics of the link mechanism, as analytically expressed by the second term of the equation, become less pronounced. In Fig. 5 the curves designated by the letter "*a*" are plotted for the term

$$\frac{1 - n\cos\theta}{K}$$

and the curves designated by the letter "*b*" are plotted for the second term of the Equation (13), that is,

$$\frac{n\sin\theta}{\sqrt{4p^2K - K^2}}$$

The selection of parts slightly larger than the driver, $m = p = 9/8$, shows a reduction in the effect of the second term of the equation on the basic curve as against that of the foregoing cases. The three points to remember in the case of equal follower and link are: (1) values of $m = p$ greater than one produce smoother resultant curves; (2) maximum ratios equal those of the slotted arm mechanism; (3) with increasing m and p values, the minimum ratio approaches that of the slotted arm.

The fourth and final case, that of equal cranks and link, is of special importance. As is shown in Figs. 2, 3 and 5, it furnishes a basis of comparison for each of the other forms. Although not yielding as smooth a velocity diagram as those which may be obtained through other combinations of lengths, this case offers many advantages. The velocity ratio formula can easily be derived from that of Case III, by putting $p = 1$, and is

$$\text{Ratio}_4 = \frac{1 - n\cos\theta}{K} + \frac{n\sin\theta}{\sqrt{4K - K^2}} \quad (14)$$

A summary of the preceding paragraphs discloses the following facts: The slotted arm or Whitworth mechanism produces a symmetrical velocity diagram with the maximum and minimum points 180 deg. apart. The link mechanism gives an asymmetrical diagram with maximum and minimum nearly 120 deg. from one another. For Cases III and IV, in which $m = p$, maximum speed occurs when the driving crank passes through the follower center, and the minimum ratio occurs when the driven crank passes over the driving shaft center. For any case where $p < m$, the maximum and minimum positions occur just after those for Cases III and IV, for values of $p > m$, the extreme velocities when compared with Cases III and IV occur a little earlier.

In selecting correct proportions for any drag-link mechanism it is necessary to remain within certain limits to prevent a reciprocating motion and to allow for rotation through 360 degrees.

First: The eccentricity n must be less than the length of any other member, that is, m and p must each be greater than the eccentricity.

Second: The eccentricity must be less than the sum of the cranks minus the length of the link, or $n < (1 + p - m)$. It is not to be assumed that any combination lying within these limits is recommended for use.

To illustrate these facts further and those previously enumerated a number of polar diagrams, Figs. 6, 7 and 8 are presented. These graphs show the location of crankshaft centers for any desired velocity ratio, the origin 0 being the follower center. The curves were constructed by locating the center of the driving crank at the point where it produced the maximum velocity of the follower crank when the latter assumed a vertically downward position. To simplify the selection of these points, the maximum velocity position was considered always to occur when the driving crank center passed through the driven center $(\theta = 0)$. This rule is correct in the majority of cases; in those instances where it fails to hold true, variations are so slight they may be overlooked. Figs. 6, 7 and 8, represent the plots for Cases I, II and III respectively, with Case IV shown on each diagram for purposes of reference.

The fact that the maximum velocity ratio usually occurs when θ equals zero is evident from Figs. 2, 3 and 5; and that in those cases where the maximum is slightly displaced, the value of R when θ equals zero is approximately equal to the maximum. Proceeding on this basis, θ was set equal to zero in Equation (10), and a general formula for the maximum ratio R_{max} was obtained. This formula,

$$R_{max} = \frac{1}{1 - n}$$

which can also be found by examining Fig. 9, suggests a simple method of finding the eccentricity for any desired speed ratio. Through the use of the formula, the values of the maximum ratios were inserted in the polar diagrams, enabling an immediate selection

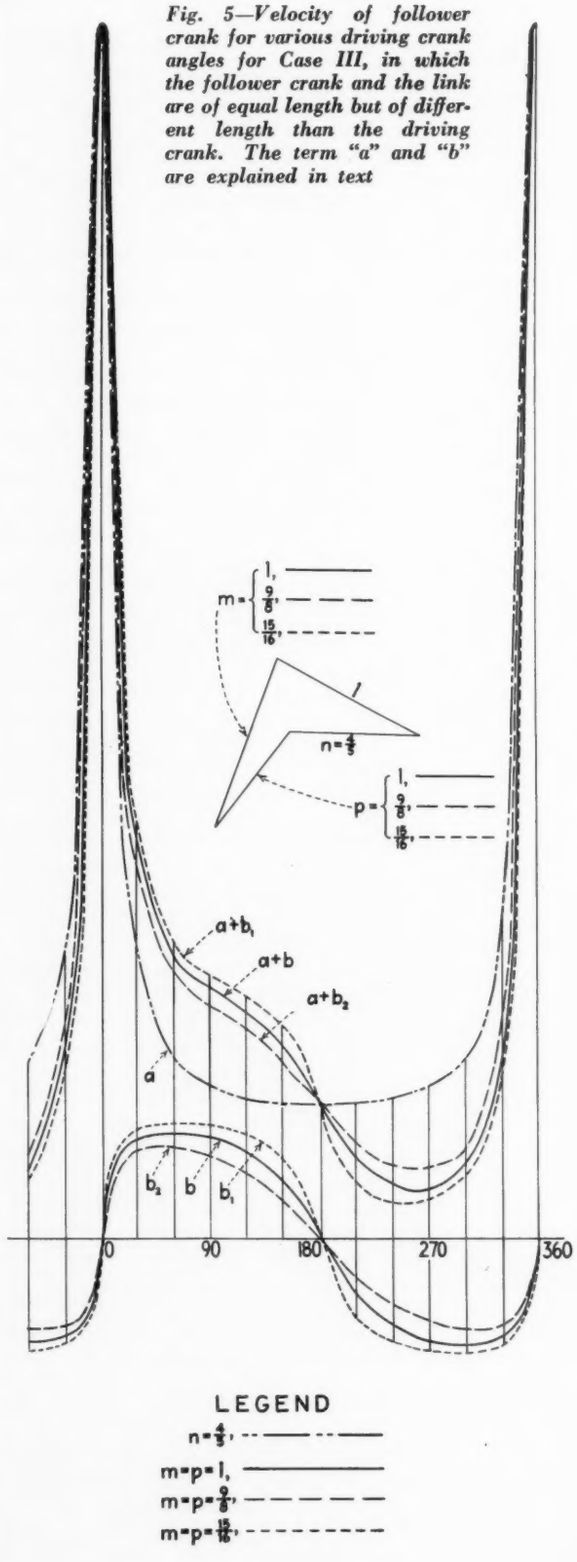


Fig. 5—Velocity of follower crank for various driving crank angles for Case III, in which the follower crank and the link are of equal length but of different length than the driving crank. The term "a" and "b" are explained in text

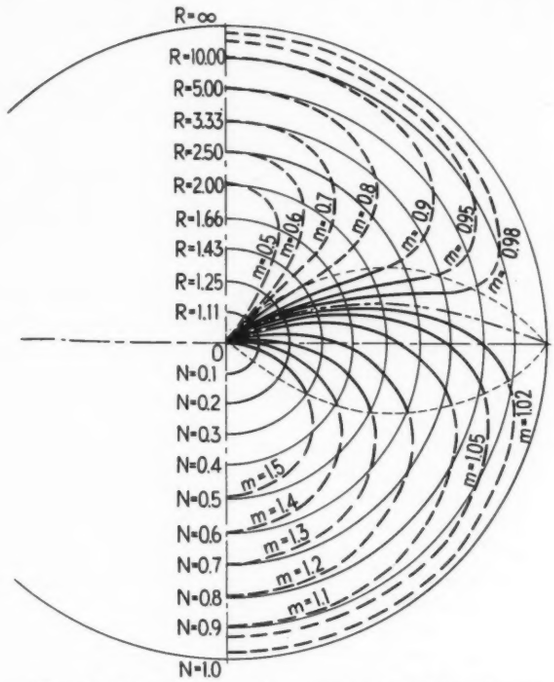


Fig. 6—Polar diagram for Case I, in which the follower crank and the driving crank are of equal length. The curves are loci of the driving crank centers for various speed ratios, the radial distance from the origin "O" to the particular point on curve is the eccentricity

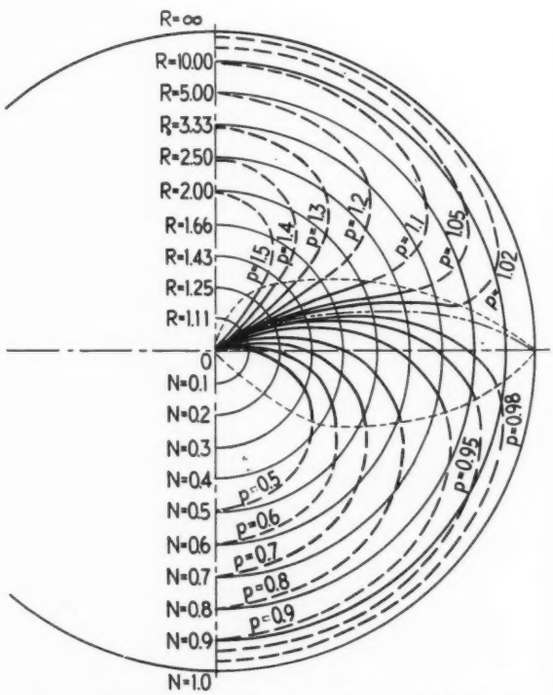


Fig. 7—Polar diagram for Case II, in which the link and driving crank are of equal length. The curves are loci of the driving crank centers for various speed ratios, the radial distance from the origin "O" to the particular point on curve is the eccentricity

of the correct eccentricity for any required maximum velocity.

The areas lying within the broken lines of Figs. 6, 7 and 8 designate eccentricities the use of which with the corresponding link proportions will give the most satisfactory operating characteristics. These combinations prevent excessive angularity between members, resulting in the elimination of high radial forces and vibration.

The polar diagrams, Figs. 6, 7 and 8, indicate a means of obtaining a set of maximum speed ratios by adjusting the drive crank center along one of the curves. All of the curves within the recommended areas may be closely approximated by circular arcs. This means that the adjustment may be made about a gear center, usually a driving pinion center. In many instances, this will lead to important structural simplification. In the case of the crank and slotted arm mechanism, straight line adjustment is necessary to bring maximum values together. This is usually accomplished through the use of a sliding member, which is eliminated by the link mechanism.

It is possible to obtain the equations of the polar curves which are the loci of the drive crank centers for maximum speed ratios. The locus of these centers for the case where $m = p$, and in which the maximum velocity always occurs when the drive crank coincides with the line of centers, is a portion of a limacon, in which the locus of the driving center is $n = 1 - 2\,p \sin \phi$.

When $m \neq p$, the equation becomes more complicated, and its value is of insufficient importance to warrant its inclusion here.

The question of choosing one of the several cases to suit operating requirements now arises. Cases I and II provide the velocity diagrams, Figs. 2 and 3, which should prove desirable in machines where relatively low maximum ratios, generally R_{max} less than 5, are necessary. Their adjusting arcs of small radius may prove favorable under many circumstances.

For higher speeds, where R_{max} is greater than 5, Cases III and IV are preferable. These are the only proportions which would provide a theoretically infinite maximum velocity ratio. Figs. 6, 7 and 8 leave no doubt on this point, especially when the manner of adjustment is examined. The curves in Fig. 8 can be closely approximated by circular arcs. An objection to the use of equal link and follower crank where $m = p > 1$ is the knowledge that the advantages of this arrangement over Case IV do not become pronounced until $m = p > 2$. If the driving crank size is to be maintained, this leads to an unnecessary increase in the length of the other parts together with the subsequent demand for greater operating space.

Case IV provides the best set of characteristics over the largest range of ratios and is recommended for general use, unless, of course, special conditions dictate the choice of other proportions. The lack of symmetry in the velocity diagram is not detrimental; the cranks can be driven in the direction which causes the steeper slope of the curve to occur during deceleration, that is, to place the link under compression while driving, rather than in tension. In this way, the friction forces within the mechanism will aid the higher deceleration while working against the lower acceleration and produce a resultant having a symmetry which is greater than that of the more symmetrical-appearing velocity diagram. When working with high ratios it is important that all parts subjected to variable velocities be made as light as possible to cut down inertia forces.

Allowable accelerations are dependent upon the individual design especially with regard to minimum overhang of cranks; the weight of the driven parts; the rigidity of the various members and their method of mounting; the friction resistance of the bearings; the speed at which the machine is operated; and the type of work to be done. Each application will have to be studied by itself in this respect, and no definite rules can be given which will apply to all designs.

The possibilities for incorporating link mechanisms in the design of a machine to improve its performance are numerous. Skillful and experienced handling of the many combinations of cranks and links will permit a greater flexibility in design and will result in the simplification of the machine.

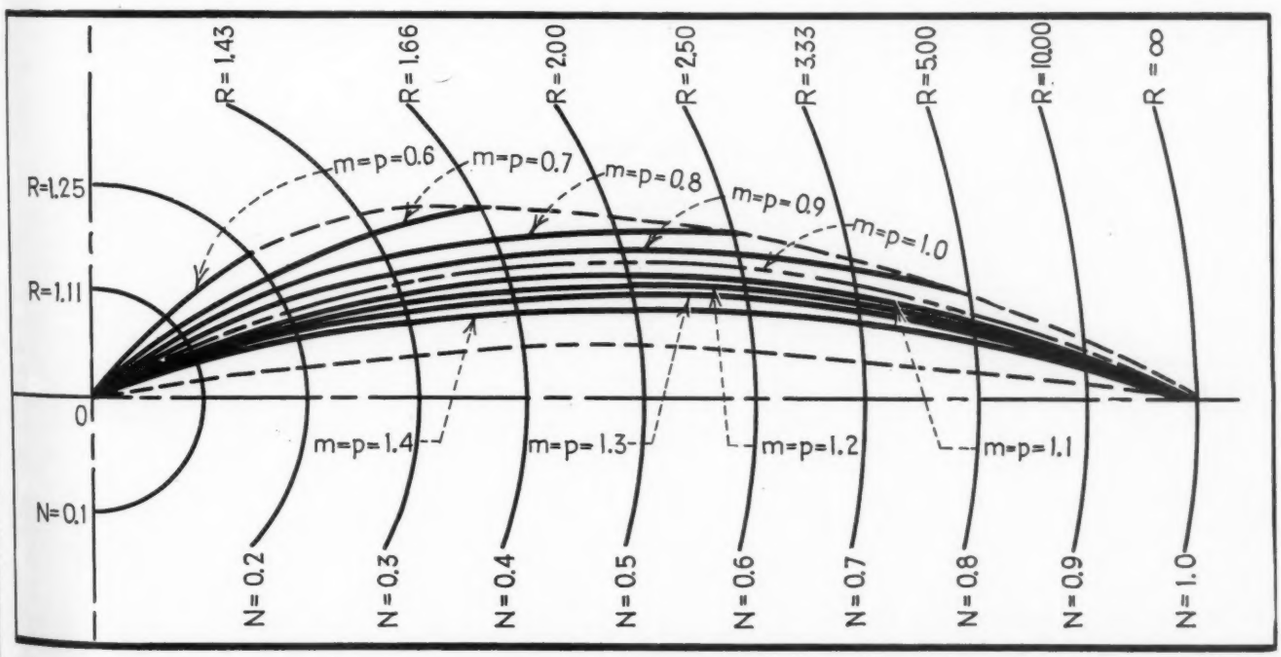


Fig. 8—Polar diagram for Case III, in which the follower crank and the link are of equal length, but of different length than the driving crank. The curves are loci of the driving crank centers for various speed ratios, the radial distance from the origin "O" to the particular point on the curve is the eccentricity

COMPOSITE MEMBERS

How to Find the Stress in Each Material

WILLIAM I. POWELL, *Assistant Mechanical Engineer,*
Board of Water Supply of the City of New York

WITH the use of clad material becoming more general in industry, the question of stresses in composite members made up of two different materials is being encountered by more and more designers. Although many authors of engineering texts on strength of materials have treated this subject to a greater or less extent, a clear restatement of the solutions of a few fundamental cases may contribute to a better understanding to those confronted with the problem.

For simplicity sake, four cases will be chosen, two with direct stresses, that is, either pure tension or pure compression, and two cases involving bending stresses.

Case I

This is the case of a tension member or of a compression member in which the ratio of the length to the least cross-sectional dimension is less than 6. It is beyond the scope of this analysis to derive suitable stress formulas for a composite column. Those interested will find that the reasoning behind the following four cases can be applied through derivations using some of the many column formulas extant to the solution of stress in such a member.

A_1 and A_2 = respective areas
e_1 and e_2 = respective changes in length of parts 1 and 2
P_{1a} and P_{2a} = actual total loads on A_1 and A_2 respectively
$P_T = P_{1a} + P_{2a}$ = total actual load carried by composite section
P_{1m} and P_{2m} = allowable maximum loads sustained on parts 1 and 2 separately
S_1 and S_2 = allowable unit stresses on A_1 and A_2
S_{1a} and S_{2a} = actual unit stresses on A_1 and A_2
E_1 and E_2 = moduli of elasticity of the two parts
$P_{1m} = A_1 S_1$ and $P_{2m} = A_2 S_2$
also, according to Hooke's law, if S_{1a} and S_{2a} are within the proportional limits of the materials then,
$P_{1a} = A_1 E_1 e_1$ and $P_{2a} = A_2 E_2 e_2$

In order that the junction of parts 1 and 2 not be ruptured e_1 must equal e_2

since $e_1 = \dfrac{P_{1a}}{A_1 E_1}$ and $e_2 = \dfrac{P_{2a}}{A_2 E_2}$

then $\dfrac{P_{1a}}{A_1 E_1} = \dfrac{P_{2a}}{A_2 E_2}$, from which

$P_{2a} = P_{1a} \dfrac{A_2 E_2}{A_1 E_1}$ and $P_{1a} = P_{2a} \dfrac{A_1 E_1}{A_2 E_2}$

As a problem consider,

$A_1 =$ sq. in. $\quad A_2 = 4$ sq. in.
$S_1 = 12{,}000$ lb./sq. in. $\quad E_1 = 24 \times 10^6$
$S_2 = 8{,}000$ lb./sq. in. $\quad E_2 = 12 \times 10^6$

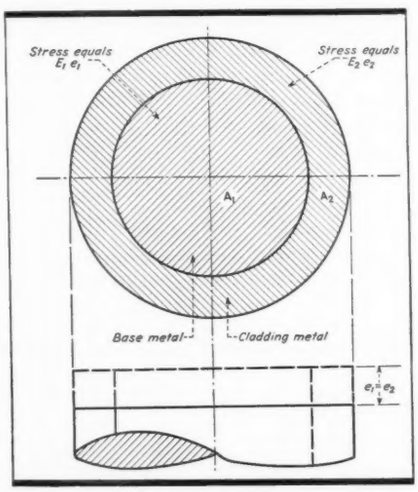


Fig. 1—In a composite bar under tension, the change in length must be equal for the two elements, and within the proportional limits, the stress in each material will be the elongation times the elastic modulus of the material

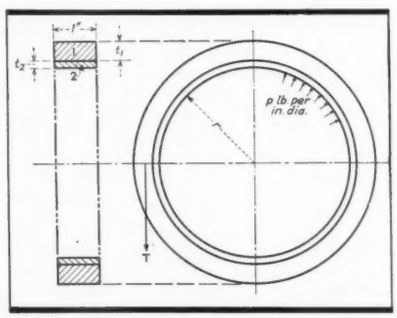


Fig. 2—In a cylindrical vessel subjected to internal pressure, membrane stresses have the same relationships as stresses in a bar under tension

Then $P_{1m} = 6 \times 12{,}000 = 72{,}000$ lb.
and $P_{2m} = 4 \times 8{,}000 = 32{,}000$ lb.
Assuming A_1 stressed to maximum,

$$P_{2a} = 72{,}000 \times \frac{4 \times 12 \times 10^6}{6 \times 24 \times 10^6} = 24{,}000 \text{ lb.}$$

and the actual unit stress on

$$A_2 = \frac{24{,}000}{4} = 6{,}000 \text{ lb. per sq. in.}$$

Assuming A_2 stressed to maximum,

$$P_{1a} = 32{,}000 \times \frac{6 \times 24 \times 10^6}{4 \times 12 \times 10^6}$$

$= 96{,}000$ lb., which is too high
Therefore P_m or the maximum allowable load on the combined section = 72,000 + 24,000 = 96,000 lb. and the load ratio between parts 1 and 2 is

$$\frac{P_{1a}}{P_{2a}} = \frac{72}{24} = \frac{3}{1}$$

Thus for any other total load,

$$P_{1a} = \frac{3}{4} P_T \text{ and } P_{2a} = \frac{1}{4} P_T$$

Case II

A pressure vessel under internal pressure p causing a direct system of membrane stresses without bending or discontinuity forces at the point considered. This is a variation of Case I.
$T = p\,r$ = force acting per lineal inch over the total section.

$$S_{1a} = P_{2a} \frac{t_1 E_1}{t_1 t_2 E_2} = P_{2a} \frac{E_1}{t_2 E_2}$$

and $S_{2a} = P_{1a} \dfrac{E_2}{t_1 E_1}$

Assuming
$p = 170$ lb. per sq. in. $\quad r = 60$ in.
$E_1 = 28 \times 10^6 \quad t_1 = \frac{1}{2}$ in.
$E_2 = 23 \times 10^6 \quad t_2 = \frac{1}{8}$ in.
$S_1 = 20{,}000$ lb. per sq. in.
$S_2 = 18{,}000$ lb. per sq. in.
Then $P_{1m} = 20{,}000 \times 0.5 = 10{,}000$ lb.
and letting $P_{1m} = P_{1a}$

$$S_{2a} = 10{,}000 \frac{23 \times 10^6}{0.5 \times 28 \times 10^6}$$

$= 16{,}400$ lb. per sq. in.
$P_{2a} = \frac{1}{8} \times 16{,}400 = 2{,}050$ lb.
$P_T = 10{,}000 + 2{,}050 = 12{,}050$ lb.
If $T = P_T = 170 \times 60 = 10{,}200$ lb.

$$S_{1a} = \frac{10{,}200}{12{,}050} \times 20{,}000$$

$= 16{,}900$ lb. per sq. in.

and $S_{2s} = \frac{10{,}200}{12{,}050} \times 2{,}050$

$= 1{,}730$ lb. per sq. in.

In pressure vessels where local bending moments exist such as at points of discontinuity between dished heads and adjoining cylindrical sections, stresses as determined in Case III or Case IV must be added algebraically to the membrane stresses found in Case II.

Case III

This is the case of a beam or unit section of a plate or pressure vessel which is subject to a bending moment. It consists, as shown in Fig. 3, of a central core and uniform cladding material on both sides of the member giving a symmetrical cross-section.

In the derivation of the common flexure formula $M = SI/C$ it was assumed that the material of the beam was homogeneous. The formula therefore, does not apply directly to a composite beam. However, since we are dealing with a symmetrical section and knowing that for equal elongation, by the law of proportionality $S_{1s}/S_{2s} = E_1/E_2$ we can proceed to develop a resisting moment which is the equivalent of the given bending moment. C_1, T_1, C_2, and T_2 are compressive and tensile forces acting on the respective parts of the beam.

$$n = \frac{E_2}{E_1}, \qquad I_1 = \frac{b\ d_1^3}{12},$$

$$c_1 = \frac{d_1}{2}, \qquad c_2 = \frac{d_1 + d_2}{2}$$

Referring to Fig. 3 it can be seen that

$$C_1 = T_1 = \frac{S_{1s}}{2} \times \frac{b\ d_1}{2} = \frac{b\ d_1\ S_{1s}}{4}$$

$$C_2 = T_2 = b\ d_2\ n\ S_{1s}\ \frac{1 + (c_1 + d_2)/C_1}{2}$$

Since the bending moment = resisting moment

$$M = \frac{S_{1s}\ I_1}{c_1} + \left(b\ d_2\ n\ S_{1s}\ \frac{1 + (c_1 + d_2)/c_1}{2}\right)(d_1 + d_2)$$

from which S_{1s} can be determined.

Then $S_{2s} = S_{1s}\ \frac{n\ (c_1\ d_2)}{c_1}$

To illustrate by an application we assume

$b = 4$ in., $d_1 = 6$ in., $d_2 = \frac{1}{8}$ in., $E_1 = 10^6$

$I_1 = \frac{4 \times 6^3}{12} = 72$, $n = 30$, $c_1 = 3$ in.,

$M = 72{,}000$ in. lb. $E_2 = 30 \times 10^6$

$$72{,}000 = \frac{72\ S_{1s}}{3} + \left(4 \times \tfrac{1}{8} \times 30 \times S_{1s}\ \frac{1 + \frac{3.125}{3}}{2}\right)(6.125)$$

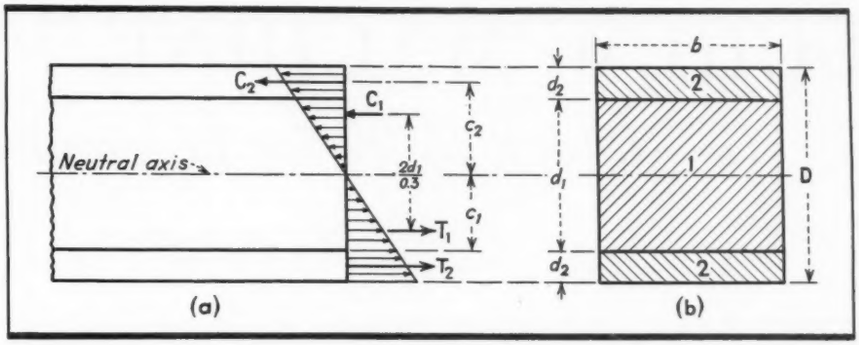


Fig. 3—In a beam with equal thickness of cladding on top and bottom, the maximum stresses in each of the materials will be proportional to the elastic modulus and the distance from the neutral axis

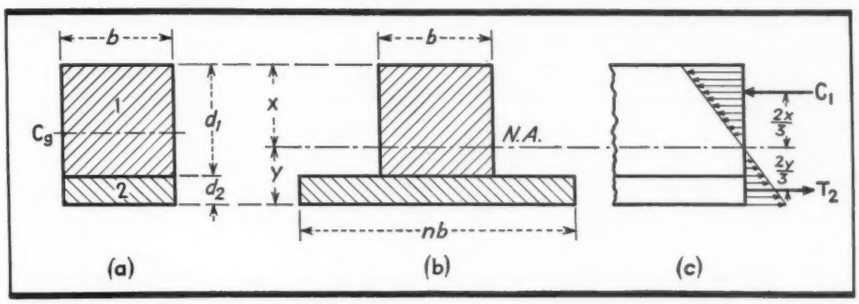


Fig. 4—With cladding on only one side of the beam, the neutral axis shifts because of the difference in the elastic modulus of the two materials. At (b) is shown the equivalent homogeneous section, the center of gravity of which locates the neutral axis of the composite beam

$72{,}000 = 24\ S_{1s} + 98.8\ S_{1s}$

giving $S_{1s} = 611$ lb. per sq. in.

$$S_{2s} = 611 \times \frac{30 \times 3.125}{3}$$

$= 19{,}094$ lb. per sq. in.

Case IV

This last case involves bending in a member clad on one side only and requires a slightly different treatment from the preceding Case III.

If we transform the section shown in Fig. 4(*a*) to one of equivalent homogeneous section it will develop that, because of the disparity between E_1 and E_2, the neutral axis will shift as in Fig. 4(*b*). It is with greater ease that the solution can be made with this transformed section; hence the theory is expanded from this equivalent figure.

I_g is the moment of inertia of Fig. 4(*b*)

$$y = \frac{bd_1\ (d_2 + d_1/2) + n\ bd_2\ (d_2/2)}{bd_1 + n\ bd_2}$$

$$n = \frac{E_2}{E_1}$$

and knowing M

$$S_{1s} = \frac{M\ x}{I\ g} \qquad S_{2s} = n\ S_{1s}\ \frac{y}{x}$$

Using an illustrative problem in which $E_1 = 10^6$, $E_2 = 30 \times 10^6$, $b = 4$ in. $d_1 = 6$ in., $d_2 = \frac{1}{8}$ in., and $M = 38{,}000$ in. lb.; then $n = 30$

$$y = \frac{4 \times 6 \times 3\frac{1}{16} + 15 \times \frac{1}{16}}{24 \times 15} = 1.94 \text{ in.}$$

$d = 4.18$ in.

$$Ig = \frac{4 \times 6^3}{12} + 24 \times 1.18^2 + 15\ (6.06 - 4.18)^2$$

$= 158.6$ in.^{4}

$$S_{1s} = \frac{38{,}000 \times 4.18}{158.6} = 1{,}000 \text{ lb. per sq. in.}$$

$$S_{2s} = 30 \times 1000 \times \frac{1.94}{4.18} = 13{,}500 \text{ lb. per sq. in.}$$

In the foregoing four examples the component materials are assumed to be isotropic in nature and the effect of any internal stresses which might be developed during the cladding process has been neglected. Improved casting technique will, it is felt, reduce this latter point to insignificance.

[Editor's Note: For the interest of those concerned, a succeeding article in the June number will treat a subject about which little has been written in most textbooks, namely, the problem presented by the stressing non-isotropic materials such as brasses and bronzes. The presentation may be considered well-timed, in view of the increasing development and use of many alloys of this character.]

SCREW MACHINE PRODUCTS

II—Design Rules for Economical Production

D. H. MONTGOMERY
New Britain-Gridley Machine Company

SOME of the design factors that affect economical production of products on screw machines, such as machinability and properties of bar and tubular materials, were discussed in Part I of this article, which appeared in the March number of PRODUCT ENGINEERING. Design considerations were also included that involved dimensional tolerances, cylindrical and tapered surfaces, and thread specifications.

Other rules that should be kept in mind when designing parts that are to be made on screw machines are listed in Table III. The rules are general and conform to common practice, hence when applied to any specific design they may be subject to exceptions, or influenced by further qualifications and considerations.

When a design calls for a shoulder or flange on an outside contour, a screw machine operation can be saved by using a shoulder or flange that makes a slight taper with a plane normal to the axis. Shoulders, like that shown at *B* in Fig. 3 and at *F* in Fig. 6, can be machined perfectly square, but to do this extra operations are required, especially on steel.

To form a shoulder the forming tool must have its side face relieved with a clearance. Even when relieved the tool may drag and not cut smoothly. In consequence, it is preferable to have the face of the shoulder if narrow make a minimum angle of 1 deg., or 89 deg. with the axis as shown at *A* in Fig. 3; if the face of the shoulder be wide the angle should not be less than 3 deg. in order to provide the forming tool with adequate clearance to obtain required smoothness of cut. Unless the drawing specifically indicates that the shoulder must be square, the producer may if tolerances permit turn the shoulder with a slight angle. If the face has to be undercut, a tool other than a forming tool has to be used, unless the forming tool can be fed in at an angle by means of a special attachment.

Rounded or broken outside corners, like tapered shoulders because of tool shape considerations, can be produced in fewer operations than can square corners. Because of the difficulty encountered in grinding forming tools with sharp square inner corners, small fillets are provided. The outside corners of pieces turned with forming tools, therefore, are slightly rounded or chamfered as shown at *C* in Fig. 3, the minimum radius is usually about 0.005 in. Broken corners are generally advantageous in that burrs are less likely to occur, assembly is also facilitated. Square corners on parts can be made with other tools but require an extra operation. Unless square corners are desired, the drawing at appropriate points should be marked "break corners."

Rather than square out completely the bottom of a hole or an internal recess, it is cheaper to provide more thickness of material for a spot relief, as shown at *E* in Fig. 3, made by a standard drill when making the hole.

External threads that are in front of

Table III—Design Rules for Screw Machine Products

1. Stock and Material Considerations

(a) Select the material which, consistent with minimum cost, machines most readily.
(b) Use the smallest suitable diameter of stock available.
(c) Reduce length of stock required to a minimum.

2. Tool and Machining Considerations

(a) Design for machining with a minimum number of standard tools unless special ones will reduce costs.
(b) Design for a minimum number and duration of machining operations.

3. Surfaces, Shoulders, Holes

(a) Surfaces to be machined wherever possible should be surfaces of rotation.
(b) Machine surfaces only when necessary or advantageous. Reduce depth and length of cuts required to a minimum.
(c) Design external surfaces so that they can be machined with a single forming tool or with rough and finish forming tools, or by a rapid turning operation.
(d) Avoid square external edges on shoulders, corners, and holes; but if required, so indicate definitely on the drawing.
(e) Avoid square bottom holes. Use a standard drill if it will meet requirements. When a square bottom hole is required, the bottom should be relieved at the center.

4. Threads

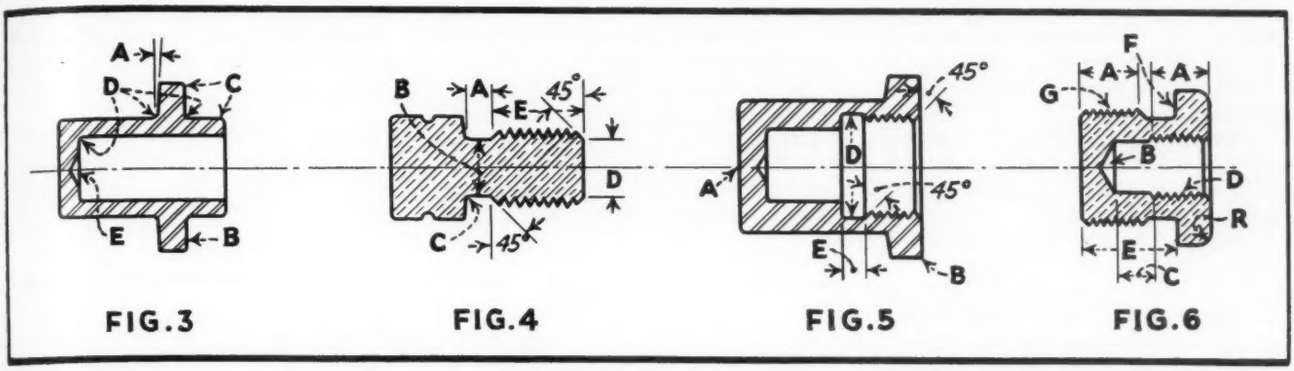
(a) Length of full threads and the percentage of full depth of tapped threads required should be indicated on drawings.
(b) Threads should not come closer than 2½ full threads to shoulders or to hole bottoms. Threads tapped close to the bottom of a blind hole or to a counterbore shoulder should be provided with a radial recess about 2½ threads in length at the end of the thread. Wherever possible avoid threads which require the use of a bottoming tap.

5. Dimensional Limits

(a) Do not specify dimensional limits which are closer than need be.

6. Surface Finish

(a) Do not specify finishes to be smoother than those normally left by the tools commonly used, unless definitely required. If special finishes are required indicate the character of extra cuts, the type of tool and where to be used; or submit a sample to indicate the finish necessary.
(b) Unless removal of burrs is necessary, do not so specify; if essential, indicate definitely where removal is necessary.
(c) If after assembly the piece is exposed to view, design it with some consideration for appearance value.



Figs. 3, 4, 5 and 6—Typical screw machine products that illustrate how tooling considerations affect economical design

shoulders are easier to machine than those which are back of shoulders or portions of larger diameters, since the threads can be cut with either self-opening or solid dies when the die can be fed over the front end of the piece. When threading to a shoulder an undercut should be provided as at *A*, Fig. 4, having a width of at least 2½ threads, and a diameter *B* about 0.010 in. smaller than the root of the thread for die run-out. A small fillet should also be provided at *C* where the undercut and shoulder join. With adjustable dies, close fits can be held on external threads. Commercial practice usually calls for Class 2 fits, with the probability that Class 3 fits will increase costs.

External threads that are in back of shoulders or flanges which have diameters larger than the thread root diameter, like that shown at *G* in Fig. 6, must either be cut with single point tools or be rolled, unless produced in secondary operations. Rolled threads are likely to be smoother and freer from fine burrs than are cut threads. Cut threads, however, meet most requirements. Threads requiring multiple starts can be produced, but they add to tooling costs because chasing tools are needed. Taper threads cost slightly more than straight threads.

Designers can facilitate the screw machine operations and tooling required to produce parts having internal threads, by observing a few important considerations. Tap drills and taps are commonly employed that produce a length of full thread that is 75 percent of the length of the drilled hole, in order to obtain sufficient depth for chip space and end clearance for standard taps. The usual minimum allowance is equal to the length of five threads beyond the straight tapped portion. A chamfer of about 45 deg. is desirable at the start of the tapped hole to facilitate centering and entering the tap.

When threads are tapped close to a shoulder or the bottom of a hole there should be a recess, like that shown at *D* in Fig. 5. The diameter of the recess should be at least 0.010 in. larger than the crest diameter of the tap; the width of the recess *E* should be equal to the length of at last three threads. A chamfer of about 45 deg. is desirable at inner end of tapped portion to avoid the formation of burrs.

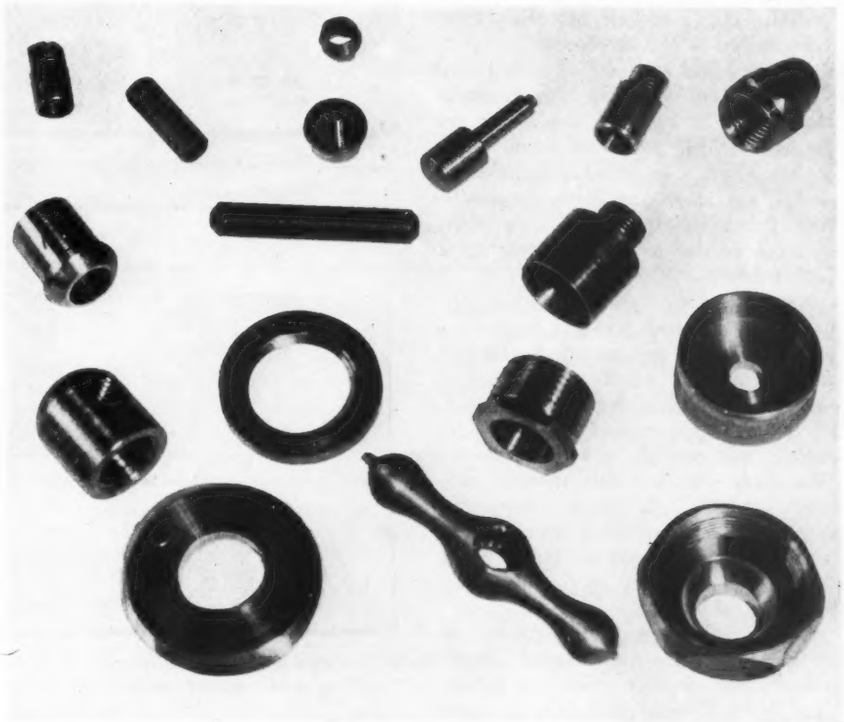


Fig. 7—Group of common parts which are produced economically on screw machines

Fits are more difficult to hold on tapped than on external threads, especially when produced with small solid taps. Class 2 fits can be held on commercial work; Class 3 fits can be held if the material taps freely.

Unfortunately, there are no generally accepted methods of designating finish or degree of smoothness on screw machine products, except by reference to the type of tool or cut in the final operation. Perhaps the best way to designate finish is to submit a sample having the required finish.

Symbols sometimes used to indicate the character of finish required are shown in Fig. 8. The letter *f* is commonly used on American drawings. Triangles are used for the same purpose, chiefly on European drawings.

The usual commercial practice calls for leaving the external surface as it is after forming or turning with the usual tool. If better smoothness is required, the outside diameters can be skived or shaved. If a still smoother finish is necessary, surfaces can be burnished with suitable rotary burnishing tools. Smoothness of interior surfaces can be improved by reaming, and sometimes interior burnishing can be done. End faces also can be burnished. Burnishing does not alter dimensions significantly,

it simply renders tool marks invisible. Contrary to some beliefs, burnishing is a good preliminary to grinding as less metal need be removed in the grinding operation.

Pieces which are to be plated usually call for polishing and buffing, the cost of these secondary operations can be minimized by obtaining surfaces as smooth as possible in the screw machine.

When a screw machine product is to be used in a location exposed to view, or if good appearance will help promote sales, care should be exercised to see that the product is well proportioned and that details which improve appearance are incorporated in the design, especially if the cost of providing these refinements is not excessive.

The punches and nail sets illustrated in Fig. 10 are screw machine products. Differences in cost are small, but the better looking products command a higher price. The chief differences in design are in the use of a decoration that "breaks-up" uninteresting surfaces, and yet at the same time performs a useful function.

Knurling, beads and grooves can be used to improve the appearance of parts. Fig. 9 shows the effect of small changes in design on the appearance of thumb screws. The design shown at *A* has the best eye appeal because of its contour and straight knurling. Design *B* may be cheaper but it looks too heavy. Design *C* is both a compromise and unsatisfactory; there are no chamfers at the edges and no land at the shoulder to avoid or to eliminate the formation of burrs.

Knurling increases the diameter of the surface, hence the drawing should indicate the diameter required before knurling, which is preferred, or state the limiting diameter after knurling. The type of knurling, the approximate number of teeth per inch, and the maximum depth of knurl should also be specified.

When knurling overlaps end faces or shoulders, the corners should be chamfered to prevent the formation of burrs. When shoulders or flanges do not interfere, knurling tools can be fed in from the end of the piece. In recesses or back of shoulders, knurling tools must be fed in from a cross slide and the width that can be knurled is limited. Knurling can often be done as a secondary operation in thread rolling machines.

Many inserts used in die-cast and in molded plastic products, have knurled surfaces which serve to anchor the insert. Straight serrations are often knurled on parts which must make a tight press fit in a mating hole, especially when the material is not as hard as the knurled part. Letters, numerals and trade marks can be sunk into parts by using a roller having raised characters tooled on its face.

In conclusion two specific designs are shown, with captions that point out factors which deserve the attention of engineers who design screw-machine products. These illustrations are included chiefly to indicate what can be done when a particular type of design is required even though somewhat difficult problems in tooling and production are involved.

[Editor's Note: This article was abstracted from a chapter of a forthcoming book edited by Herbert Chase, entitled "Designing for Quantity Production."]

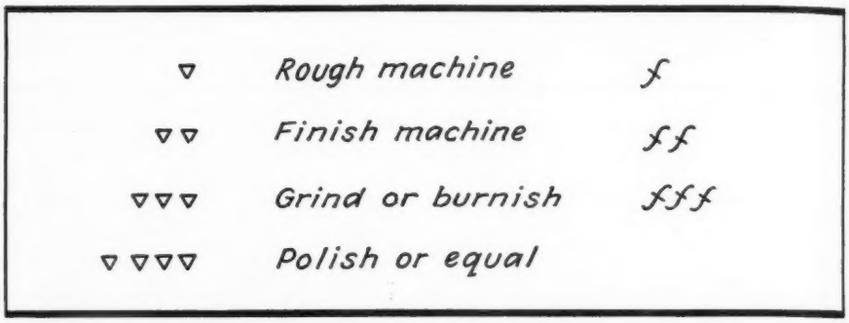


Fig. 8—Symbols used to indicate type of finish desired on screw machine products

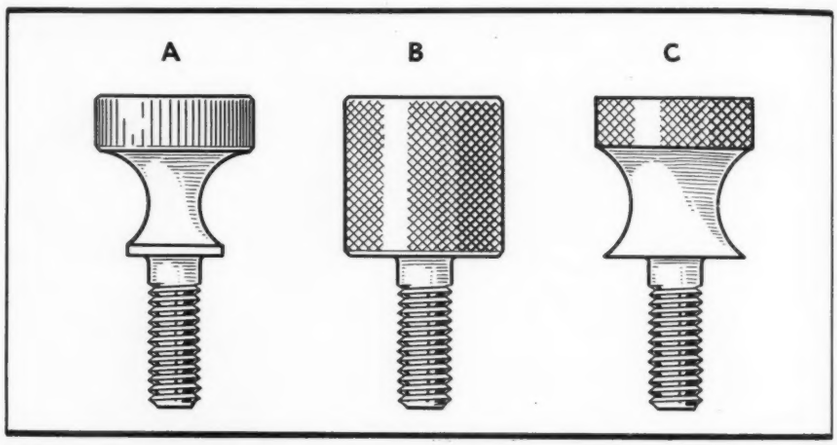


Fig. 9—Thumb screws illustrate how appearance can be enhanced by careful use of knurling and contour proportions

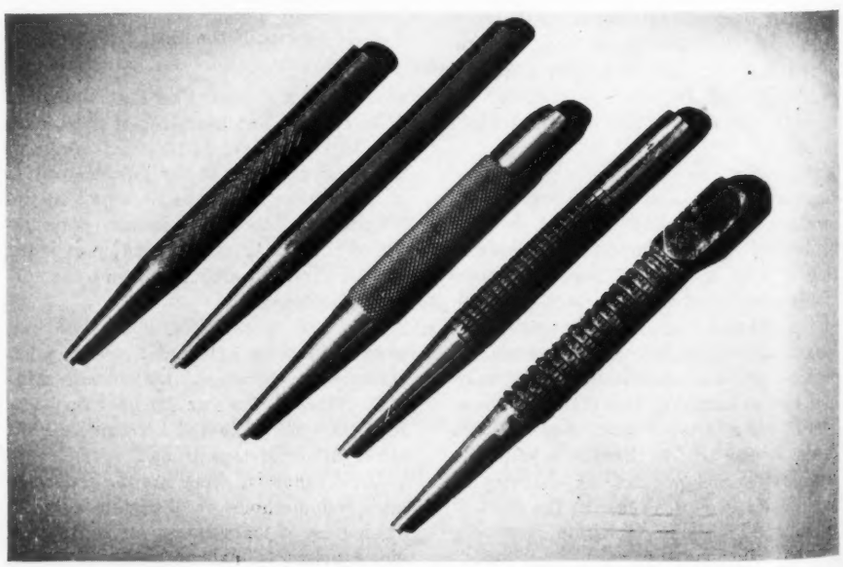
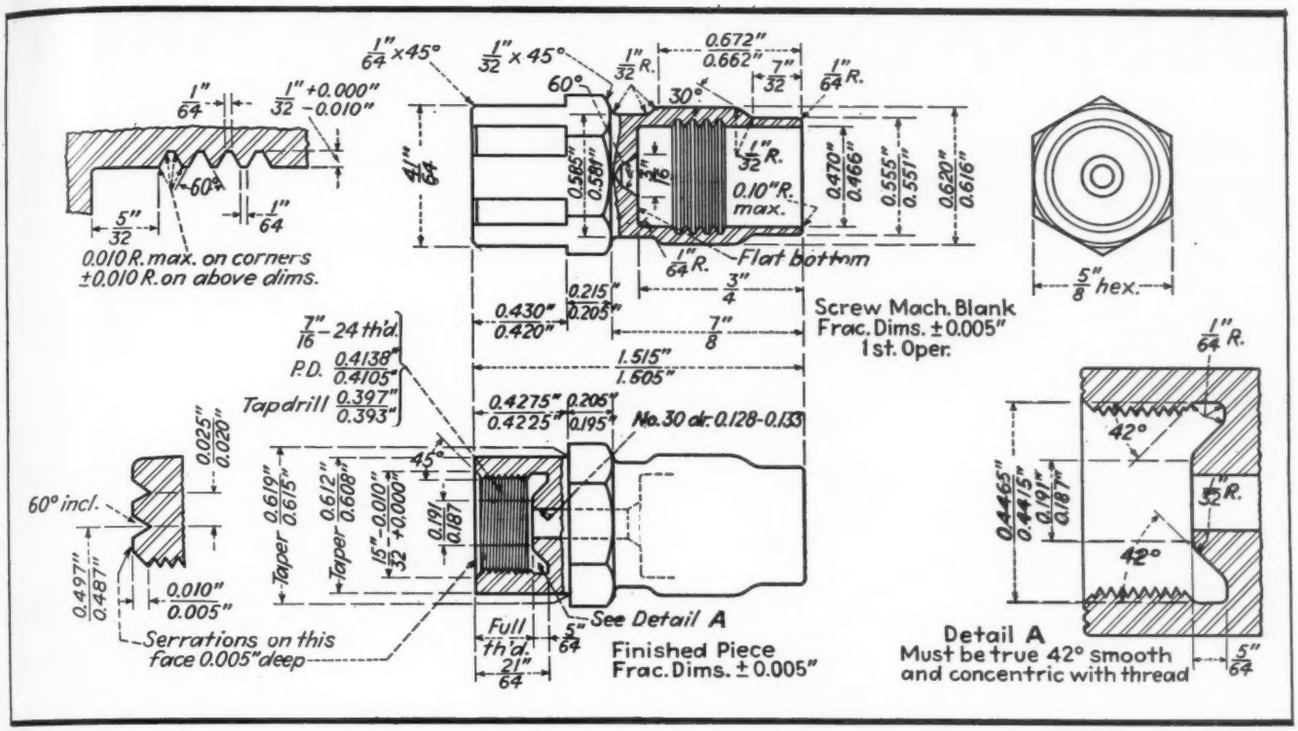
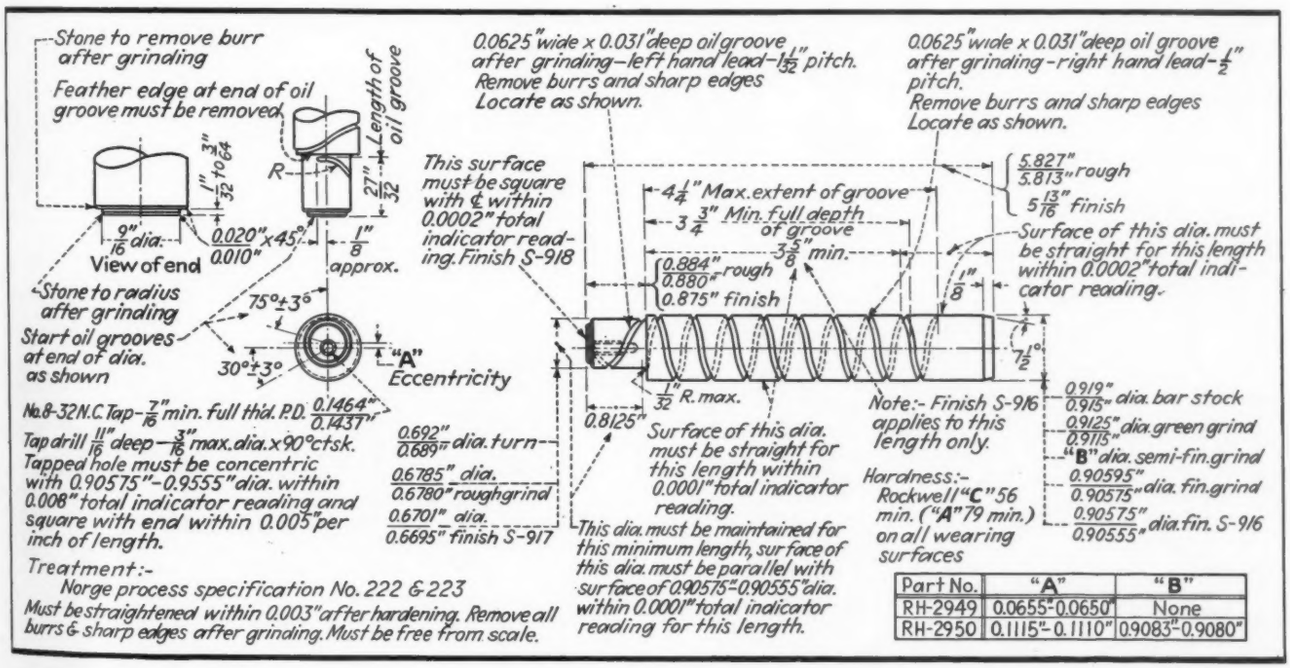


Fig. 10—Punches or nail sets produced in a screw machine. Differences in cost are small, but the better looking products command a higher price



Typical drawing for a part made on a multiple-spindle screw machine. Internal circumferential grooves in the longer hole are unusual and require special tooling, as do the production of the 42 deg. bevel at the bottom of the threaded hole and the thread relief at the inner end of this thread. Several secondary operations are required in producing this piece because of the recesses in both ends and the design of internal surfaces. Note how the required diameters are indicated for the external taper at one end, and also how the fillets and radii are designated where wanted. But there is nothing, other than dimensional limits to indicate whether shoulders must be square.


Stone to remove burr after grinding
Feather edge at end of oil groove must be removed
View of end
Stone to radius after grinding
Start oil grooves at end of dia. as shown
Length of oil groove
0.0625" wide x 0.031" deep oil groove after grinding—left hand lead—1/32" pitch.
Remove burrs and sharp edges
Locate as shown.
0.0625" wide x 0.031" deep oil groove after grinding—right hand lead—1/2" pitch.
Remove burrs and sharp edges
Locate as shown.
This surface must be square with ℄ within 0.0002" total indicator reading. Finish S-918
4 1/4" Max. extent of groove
3 3/4" Min. full depth of groove
3 5/8" min.
5.827" 5.813" rough
5 13/16" finish
Surface of this dia. must be straight for this length within 0.0002" total indicator reading.
0.884" 0.880" rough
0.875" finish
"A" Eccentricity
No. 8-32 N.C. Tap—7/16" min. full thd. P.D. 0.1464" 0.1437"
Tap drill 11/16" deep—3/16" max. dia. x 90° ctsk.
Tapped hole must be concentric with 0.9057 5"-0.9555" dia. within 0.008" total indicator reading and square with end within 0.005" per inch of length.
0.692" 0.689" dia. turn—
0.6785" dia.
0.6780" rough grind
0.6701" dia.
0.6695" finish S-917
0.8125"
Surface of this dia. must be straight for this length within 0.0001" total indicator reading.
This dia. must be maintained for this minimum length, surface of this dia. must be parallel with surface of 0.90575"-0.90555" dia. within 0.0001" total indicator reading for this length.
Note:- Finish S-916 applies to this length only.
Hardness:- Rockwell "C" 56 min. ("A" 79 min.) on all wearing surfaces
0.919" 0.915" dia. bar stock
0.9125" dia. green grind
0.9115" "B" dia. semi-fin. grind
0.90595" 0.90575" dia. fin. grind
0.90575" 0.90555" dia. fin. S-916
Treatment:-
Norge process specification No. 222 & 223
Must be straightened within 0.003" after hardening. Remove all burrs & sharp edges after grinding. Must be free from scale.

Part No.	"A"	"B"
RH-2949	0.0655"-0.0650"	None
RH-2950	0.1115"-0.1110"	0.9083"-0.9080"

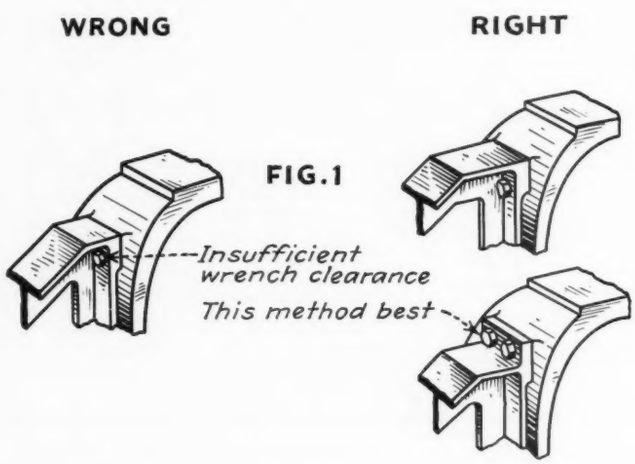
Refrigerator compressor shaft is turned with the eccentric diameter and the longer of the two oil grooves in a multiple screw machine at the rate of 1,100 pieces in eight hours. Special tooling includes the use of reciprocating rough and finish turning tools for the eccentric and of a milling cutter to form the long oil groove. The oil groove on the eccentric and the tapped hole in that end require supplementary operations, as does the precision grinding of the two diameters. The shaft is machined from S.A.E. 4620 steel, it is heat-treated after machining but before grinding. Especially noteworthy are the specific instructions on the drawing regarding tolerances and the removal of burrs. Note definite instruction for point at which to start oil grooves in reference to axis of eccentricity.

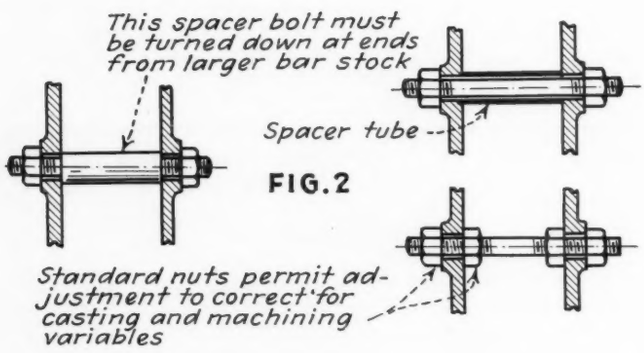
DESIGNING FOR ASSEMBLY

SKETCHES on these pages illustrate how many common machine details can be altered slightly by the product engineer in order to save unnecessary assembly work and to achieve accurate fits and

ERNEST GEIGER

alignments. This article, based on approved shop practices, finishes this series by Dr. Geiger. The rights and wrongs of "Castings and Forgings" appeared on pages 72 and 73 of PRODUCT ENGINEERING for February; "Design for Machining" appeared on pages 122 and 123 of the March number.




This spacer bolt must be turned down at ends from larger bar stock
Spacer tube
FIG.2
Standard nuts permit adjustment to correct for casting and machining variables

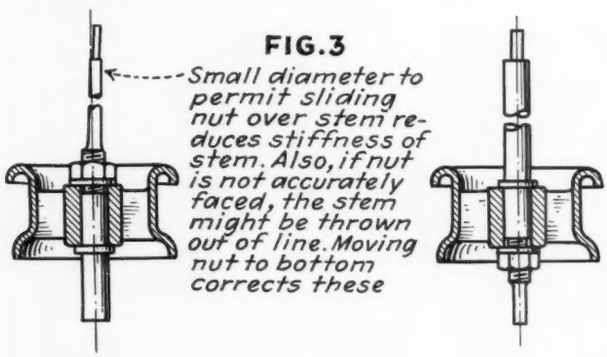
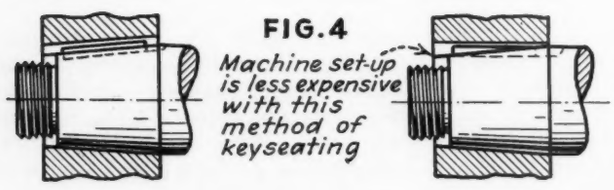
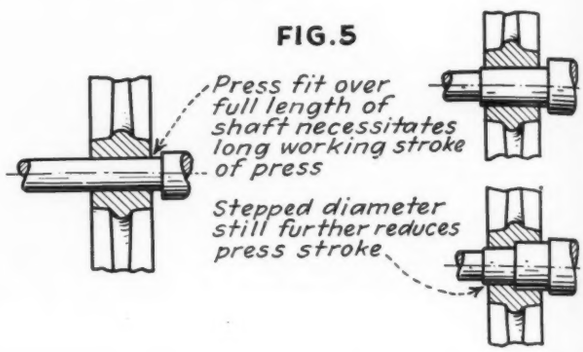
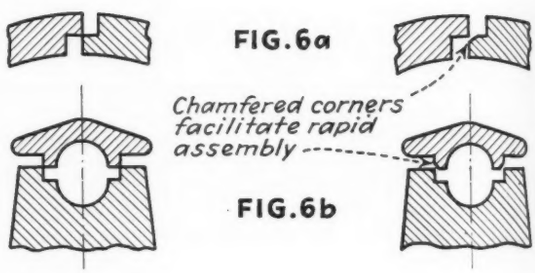


FIG.3
Small diameter to permit sliding nut over stem reduces stiffness of stem. Also, if nut is not accurately faced, the stem might be thrown out of line. Moving nut to bottom corrects these




WRONG
RIGHT
FIG.5
Press fit over full length of shaft necessitates long working stroke of press
Stepped diameter still further reduces press stroke



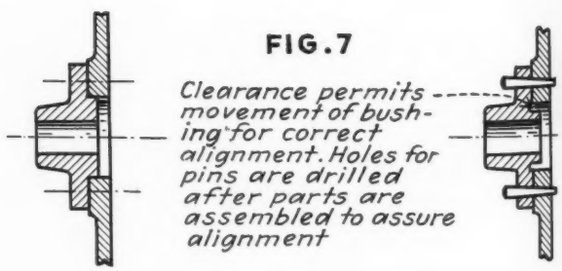
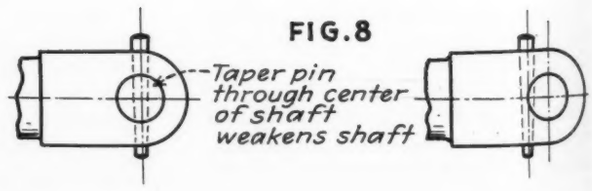


FIG.7
Clearance permits movement of bushing for correct alignment. Holes for pins are drilled after parts are assembled to assure alignment



The Rights and Wrongs of Details

WRONG RIGHT

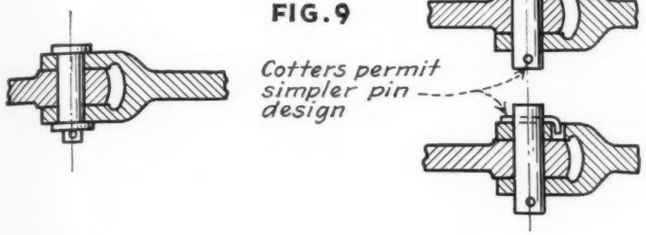


FIG. 9

Cotters permit simpler pin design

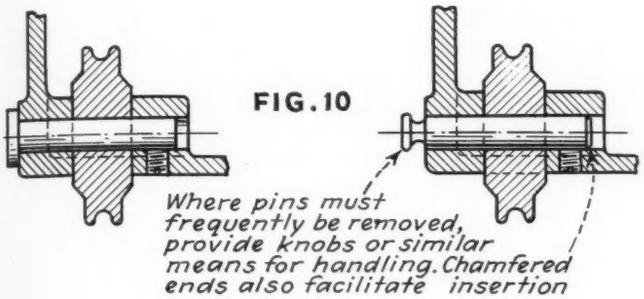


FIG. 10

Where pins must frequently be removed, provide knobs or similar means for handling. Chamfered ends also facilitate insertion

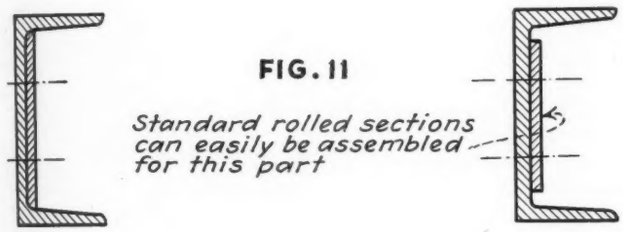


FIG. 11

Standard rolled sections can easily be assembled for this part

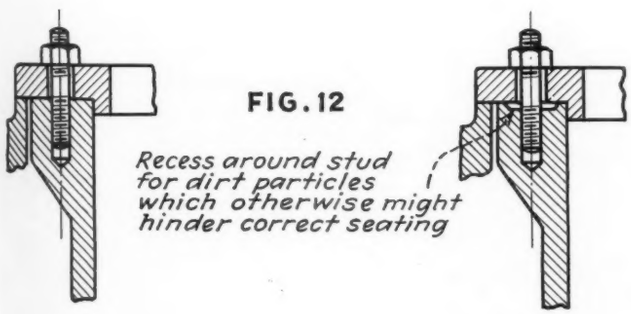


FIG. 12

Recess around stud for dirt particles which otherwise might hinder correct seating

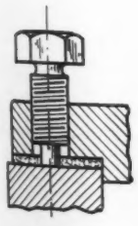
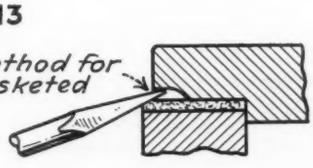


FIG. 13

Simpler method for lifting gasketed covers



WRONG RIGHT

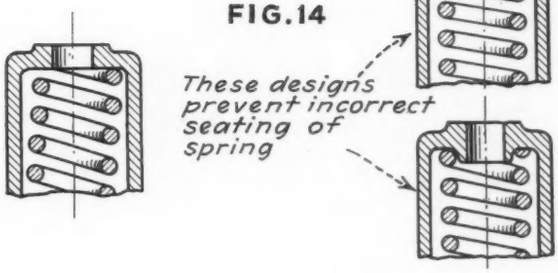


FIG. 14

These designs prevent incorrect seating of spring

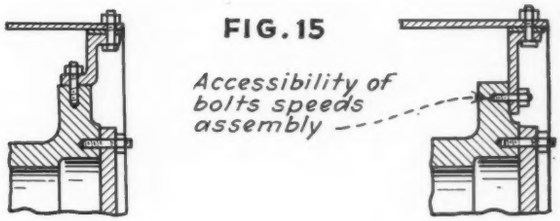


FIG. 15

Accessibility of bolts speeds assembly

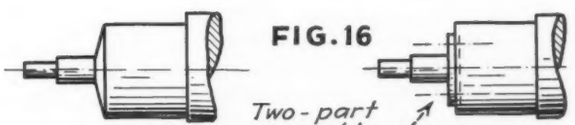


FIG. 16

Two-part assembly saves machining

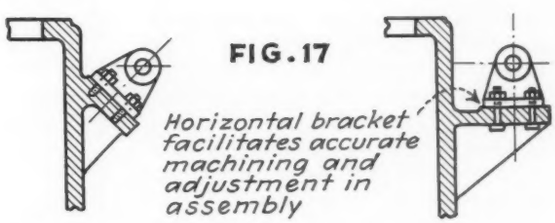


FIG. 17

Horizontal bracket facilitates accurate machining and adjustment in assembly

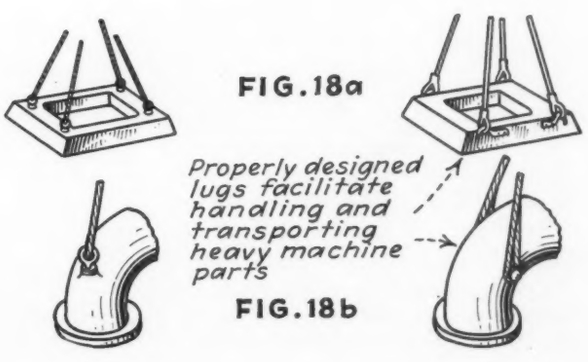


FIG. 18a

Properly designed lugs facilitate handling and transporting heavy machine parts

FIG. 18b

RUBBER-LIKE MATERIALS—II

Applications for "Synthetic" Rubbers

J. DELMONTE

VARIOUS industrial applications for the synthetic rubber-like materials, which were described in Part I of this article that appeared in the March number of Product Engineering, can be grouped into several classifications.

VIBRATION ABSORBERS—Oil or temperature has sometimes placed limits on the use of natural rubber for vibration absorption applications. The rubber-like synthetic resins make available a wide assortment of materials with varying degrees of resiliency to choose from. Materials such as plasticized polyvinyl chloride or polyvinyl alcohol possess higher damping factors than other materials, inasmuch as their rate of recovery from stress is not as rapid. Their low resilience excludes them from designs involving very rapid change of deflection with load. On the other hand, Buna rubbers, Butyl rubber and neoprene will exhibit elastic properties much more like natural rubber, having a higher proportion of resilience.

The combination of oil resistance and vibration isolation properties are useful in applications such as fuel lines and lines of hydraulic brake systems in automobiles. A length of neoprene hose is employed on the oil pressure gage line between the engine and instrument panel of an automobile because of its heat and oil resistance plus flexibility which isolates the engine vibration.

In designing machine mountings either in compression or shear, care must be observed to avoid excess creep by overloading or operating at a high temperature. In general, the load upon the mounting should not exceed that necessary to cause a 20 per cent deflection, and operating temperatures beyond 150 deg. F. are not usually recommended. Further advantages are claimed for some neoprene compounds in machine mountings because of greater damping, the larger hysteresis loops of the stress-strain curves indicate a greater energy absorbed than for natural rubber compositions of the same Durometer reading.

COATINGS FOR WOVEN FABRICS—Various woven textiles have been rendered waterproof by friction calendering them with different neoprene compositions. In this operation the neoprene is forced into the interstices of the woven fabrics by differential calendering rolls, schematically shown in Fig. 4. The properties of the finished materials are substantially those of the surface coating. Noteworthy among rubber-like materials as applied to fabrics is Koroseal which may be calendered or spread from solutions on the surface of many fabrics, increasing their gloss and rendering them waterproof. This treatment has become highly popular for wearing apparel, and has a potential utility in the industrial trade. The water-proofness is attained in the fabric without loss of flexibility, primarily because of the rubber-like properties in the applied continuous film. Koroseal is often used without fabric.

Two important applications of coated fabrics have attained much prominence recently. One is fuel cell development for airplane gasoline tanks in which a neoprene impregnated fabric bag is placed inside of the metal compartment. The unit thus continues to be leakproof under vibration, while the mechanical stresses and strain are carried by the metal structure as before.

Applications of neoprene compounds to balloon fabrics are also proving their utility inasmuch as they possess excellent resistance to sunlight. For the same thickness, they show a permeability of 1/5 to 1/6 that of natural rubber to hydrogen, helium, and carbon dioxide. The rate of diffusion is inversely proportional to the thickness. Buna, Koroseal and Thiokol coatings have also been developed for baloon fabrics.

Other calendered products include friction belting which is made for operation in atmospheres which adversely affect natural rubber. Diaphragms are often made from flat calendered sheets.

SYNTHETIC RUBBERS ON FELT—Neoprene and Thiokol compositions are being applied to felts for grease seals and gaskets. The synthetic rubber stock is milled and then calendered to various gages. These sheets are then calendered between layers of felt. Vulcanization of the laminated stock is completed under hydraulic pressure. Various layer arrangements are available, such as:

1. Felt-neoprene-felt
2. Surface of neoprene-felt-neoprene-felt
3. Felt-neoprene-felt-neoprene-felt

The second combination has been the most popular. For example, in an auto-

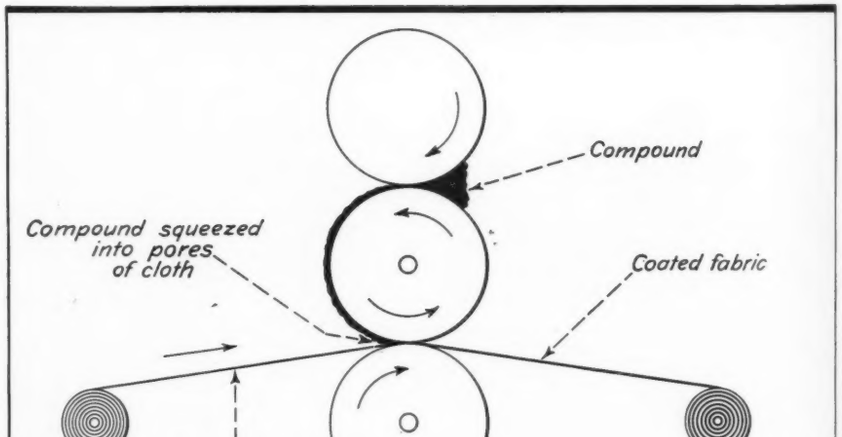


Fig. 4—Schematic arrangement of differential calendering rolls used to force compounding material into the interstices of woven fabrics

motive grease seal application, the inner layer of neoprene prevents the passage of oil, while the surface layers serve as dust guards which keep abrasive dust away from rotating parts. Fig. 5 shows a combination used with an oil filled bearing. In another type the entire surface of the felt is coated with a Thiokol composition and used as a grease seal. Under clamping pressure the Thiokol will flow and provide a good permanent seal. Thiokol with fabric facing for strength and wear resistance is used for oil slinger rings on automobile shafts.

ELECTRICAL INSULATION—Probably the best known application of synthetic rubber-like materials in the electrical industry is as insulation on electrical conductors and on cables. Much work has been done in this country and abroad using Thiokol, Buna S, Koroseal, and Oppanol B (polyisobutylene) as protective sheathings for cables. The low moisture absorption of Thiokol makes possible its use as a protective sheath for underground cables. Oil resistance, oxygen and ozone resistance, and weather exposure resistance are important requirements which are readily fulfilled by the synthetic rubber-like materials.

Numerous rubber substitutes have been developed for insulating tapes, tubings and extruded sheaths covering electrical conductors. The permanency of quality of plasticized polyvinyl chlorides, such as Flamenol, has been proven when applied directly over electrical wires.

Among the rubber-like insulating tapes and tubings the plasticized polyvinyl resins appear outstanding and fully the equal of varnished cambric in physical and electrical properties, as well as cost. They are more durable and flexible and will not become embrittled with heat or aging. They are employed in covering the electrical wiring of a number of automobiles, providing a permanent seal against weather, moisture, oil, and grease. By baking at 140 deg. C. the material can be bonded to itself to afford a continuous seal for installations where a wrapped construction is employed.

Thiokol A is best used as a non-insulating flexible jacket for cable because of its high resistance to moisture absorption and penetration. Its electrical properties limit it to low-voltage insulating purposes. Where higher physical properties and more abrasion resistance is required Thiokol DF is used as a protecting jacket for cable. It can be used from —40 to around 200 deg. F. temperature.

Plasticized polyvinyl resins like Koroseal or Irvolite are most commonly used as strip, tape or tubing which remains flexible down to —20 deg. C. and has been used under continuous operation at 80 deg. C. Being light colored and almost transparent it can be compounded in a series of bright distinguishing colors. It has good dielectric strength and resists corona cutting.

Neoprene is used in cable construction where resistance to intense sunlight, high sun temperatures, oil, and abrasion is required in addition to good electrical properties. For that reason, it has been used as protection for ignition wire, and for neon sign cables. Its resistance to acids, alkalies and chemicals has made it useful jacketing material in chemical plants and oil refineries. The new Butyl rubbers will likely see service in chemical plants handling acids and alkalies.

Perbunan is inferior to Buna S and to rubber in electrical properties and is best used as tough solvent and oil resistant jackets over insulation.

Vistanex Medium is useful on cables because it remains flexible at —50 deg. C. and resists solvents containing alcohols, esters, ketones; vegetable and animal fats; and most common chemicals except chlorine and bromine. One disadvantage is that it is difficult to get the material tough enough to withstand handling and transportation.

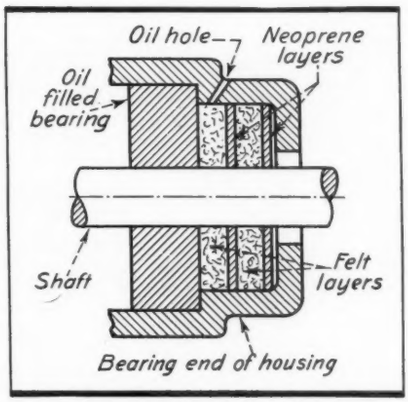


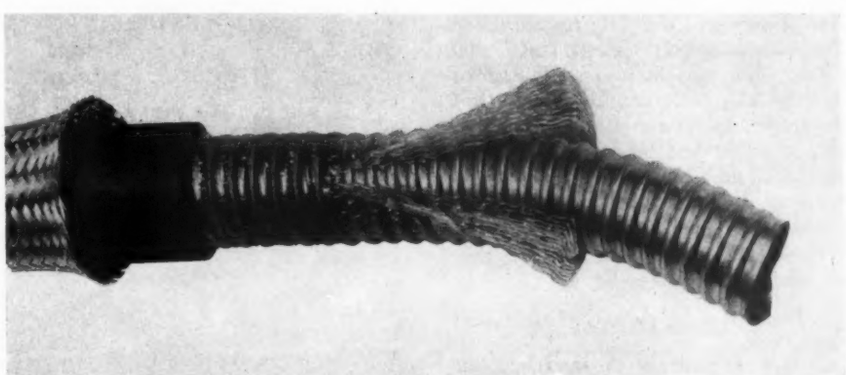
Fig. 5—Oil seal, made up of alternate layers of felt and neoprene, on shaft with oil filled bearing

CHEMICAL EQUIPMENT—In the field of chemical engineering there are applications of rubber products which serve not only to relieve mechanical strain but also to offer resistance to the corrosive action of chemicals. Prominent in these applications are plasticized polyvinyl chloride, polyvinyl alcohol, and polyisobutylene used either as a lining or molded part in a complete assembly.

1. Hose and conduits—Neoprene, Perbunan, and Thiokol, since they swell less than natural rubber, are commonly used for hose lining for handling gasoline and oils.

A new type of hose developed for oil, gasoline, and air lines of diesel engines has been prepared by Chicago Metal Hose Corporation. As illustrated in Fig. 6, a laminated cellulosic foil sheet over a flexible core, is followed in turn by a neoprene sheathing for external protection. Complete protection against gas and oils is provided by the cellulose derivative. Extruded conduits and conduit fittings are also available in Germany, prepared from Nipolam, which are particularly useful for handling corrosive chemicals.

2. Tank Linings—With the exception of oxidizing acids and hot, concentrated acid solutions synthetic rubber-like materials are definitely suited for protective linings in tanks. Only the polyvinyl chlorides are suitable for handling oxidizing acids. Various grades ranging from the soft to the hard are used for this purpose, the harder forms often being employed as bearing surfaces. As for alkalies, Thiokol is not recommended for concentrated solutions. Among the Buna rubbers, grade S is popular for handling chemicals and acids. A strong bonding cement is made by adding Thiokol to sulphur cements,



Chicago Metal Hose Corporation

Fig. 6—In this type of hose a laminated cellulosic foil sheet over a flexible core is followed in turn by a neoprene sheathing for external protection. Hose was developed for oil, gasoline and air lines of diesel engines

suitable for bonding refractory bricks in the construction of acid handling tanks.

Some mention should also be made of another useful form of rubber-like plasticized polyvinyl chloride as protective coatings for electroplating racks. Resisting alkalies and acids, Korolac has been a very useful material for coating plating racks. It is applied while hot, around 220 deg. F., to the part to be coated, and when dried for several hours will change from the jelly-like consistency it was at first, to a firm adherent coating. This type of coating works out best when it is shrunk around an article or metal part, since it tends to pull away from inside corners thus tending to curve around rather than fitting in snugly. Neoprene has also been used as a protective coating for electroplating racks.

3. Miscellaneous molded parts—Molded parts of synthetic rubber-like materials are widely employed in handling chemicals. In refrigerators, for example, neoprene parts handle refrigerants, though more satisfactory results are reported for freon than for methyl chloride. Polyvinyl alcohol has high resistance to solvents and has been used successfully to handle refrigerants. However, because of its basic chemistry it dissolves where water, alcohol or glycerine is present. Diaphragms such as those employed in airplane engine carburetors for inverted flight, molded grease seal cups in fuel pumps and paint sprayers, are a few of the many applications of synthetic rubber-like materials for molded parts.

Parts for Machine Drives—Several types of flexible couplings for machine drives have been molded from synthetic rubbers where natural rubber couplings were shortlived because of the presence of gasoline or oils. One of these was a resilient disk for the magneto drive in an aircraft engine. The resilient coupling damps out slight variations in rotational velocity between the engine and the magneto. In another machine design, a neoprene coupling transmits the drive of a centrifuge operating at 19,000 r.p.m. Two pins fit into holes molded into the body of the coupling. V-belts of neoprene used on a group of sewing machines maintained tension for a longer period of time than rubber ones previously used which failed because of the presence of oil.

Gaskets prepared from synthetic rubber-like materials may be:

1. Punched from pure synthetic rubber sheet stock
2. Molded from synthetic rubber-like materials, with or without fillers, such as cork
3. Punched from sheeting mechanically reinforced with woven fabrics

Whether natural rubber or a rubber substitute is used depends on the operating conditions which may or may not adversely affect physical properties. The plastic materials such as Thiokol which have a tendency to flow under pressure fill up any openings in a joint and make tighter seals in assemblies. However, there are many applications where the superior physical properties of the neoprene or Buna rubbers are to be preferred. Typical of these are molded packings in hydraulic presses where exceptionally good results are obtained with these materials.

Bonding to Metals—Among the most satisfactory adhesives for bonding neoprene and Buna products to metal surfaces are the chlorinated rubber cements. Some, such as Tornesit, have been successfully used to bond neoprene to aluminum, brass, steel, 18–8 stainless steel and copper. A new powdered Thiokol can be applied to metals with a compressed air and gas gun which fuses the material and causes it to flow together in a continuous film on metal surfaces.

This review has only touched upon the many highlights of the application of synthetic rubbers and rubber-like materials which are of direct interest to machine and product design. There are a number of applications which are not covered, but by studying the fundamental properties of the various materials a general design procedure for best applying the materials will be apparent. An important factor to remember in the application of synthetic rubber materials is that each one lends itself to wide variation in certain of its properties so that almost any design requirement can be met by adjustment of the mixture.

It is of interest to note the recent efforts of the rubber manufacturers to develop self-sufficiency insofar as the rubber supply is concerned. The problem is of much importance, about 600,000 tons of rubber are consumed annually in the United States. Reserve stocks are not more than enough for several months. Nevertheless, neoprene production is approaching 1,500,000 lb. per month, six times that which was produced in 1939. Reliable figures from other manufacturers are not available, though it must be realized that it is difficult to obtain these, because of the transient nature of many present developments.

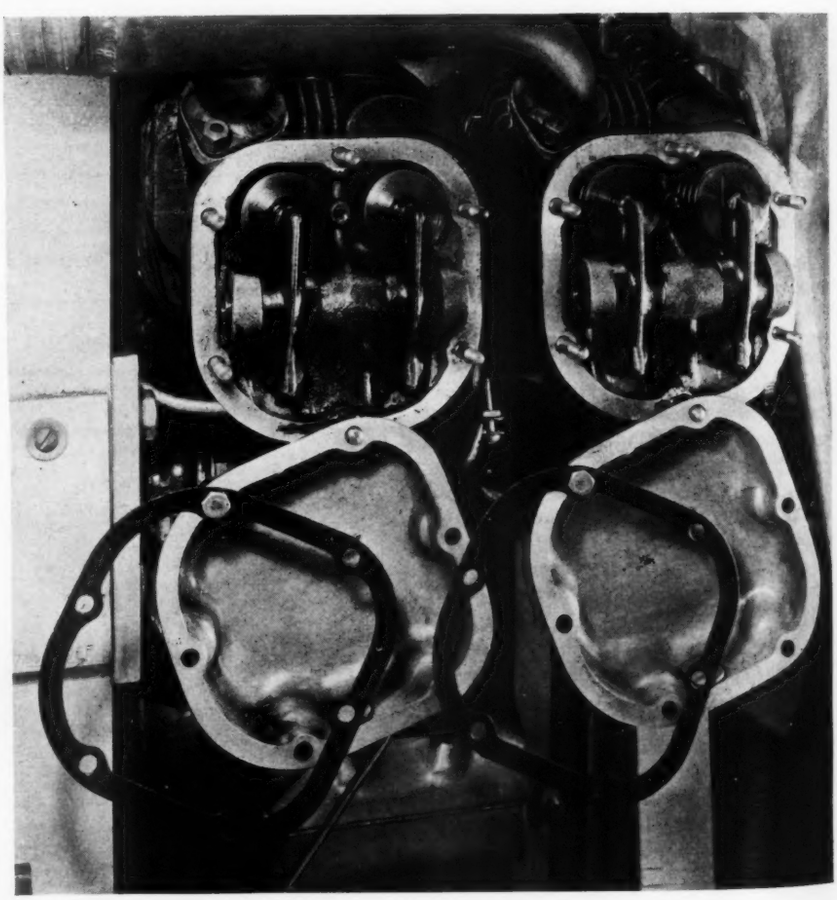


Fig. 7—Gaskets, to withstand the effects of heat and oxygen, made from neoprene cement coated asbestos for a rocker box cover on an airplane engine

CONTAINER DIMENSIONS

Minimum Area of Material for Given Volume

CHARLES E. FREDERICKSON
Johnson & Johnson

SINCE the cost of any container is a direct function of the amount of material necessary to completely inclose the desired volume, it is possible to set up a mathematical relation between the three dimensions: length, width, and depth, for the least possible surface area of the container. This is determined through application of the maxima and minima theory of differential calculus. Although the following analysis is based on the dimension of a folding container, by simple analogy and slight changes in the formulas, the method may be utilized for determining minimum amounts of material for built-up housings.

l = length of container (inside dimension)
w = width of container (inside dimension)
d = depth of container (inside dimension)
V = volume of container
A = area of material in container

Width w always refers to the dimension across the outside meeting flaps.. Hence, each of the outside flaps, is $\frac{1}{2}\ w$ in width. The length l is always the dimension along which the outside meeting flaps open. The depth d is always the remaining dimension, common to all four sides in which there is no opening.

Hence $V = lwd$

$$A = 2wd+2ld+4\frac{w}{2}\,l+4\frac{w}{2}\,w \qquad (1)$$

$$A = 2wd+2ld+2wl+2\,w^2 \qquad (2)$$

As explained previously, for lowest cost, obtain the minimum value of A. By rearranging (1) we obtain

$$w = \frac{V}{ld} \qquad (3)$$

Substituting this value of w in (2)

$$A = \frac{2V}{l} + 2ld + \frac{2V}{d} + \frac{2V^2}{l^2d^2} \qquad (4)$$

By partial differentiation

$$\frac{\partial A}{\partial l} = -\frac{2V}{l^2} + 2d - \frac{4V^2}{d^2l^3} \qquad (5)$$

$$\frac{\partial A}{\partial d} = 2l - \frac{2V}{d^2} - \frac{4V^2}{d^3l^2} \qquad (6)$$

To determine minimum values, set the first derivatives equal to zero. Hence, (5) and (6) become equal

$$-\frac{2V}{l^2} + 2d - \frac{4V^2}{d^2l^3} = 2l - \frac{2V}{d^2} - \frac{4V^2}{d^3l^2}$$

$$d - \frac{V}{l^2} - \frac{2V^2}{d^2l^3} = l - \frac{V}{d^2} - \frac{2V^2}{d^3l^2}$$

$$d = l \qquad (7)$$

Substituting (7) in (6),

$$2l - \frac{2V}{l^2} - \frac{4V^2}{l^5} = 0$$

$$l^5 - Vl^3 - 2V^2 = 0 \qquad (8)$$

Factoring,

$$l = \sqrt[3]{-V} \quad \text{(unreal)}$$

$$l = \sqrt[3]{2V} \quad \text{(real)} \qquad (9)$$

from (7) $d = l = \sqrt[3]{2V}$
" (1) $2V = 2\,(lwd)$
" (7) $2V = 2\,(wl^2)$
" (9) $l^3 = 2wl^2$

$$l = 2w \qquad (10)$$

Substituting (10) in (9),

$$w = \sqrt[3]{\frac{V}{4}} \qquad (11)$$

These, then, are the mathematical relations between the three dimensions which will give minimum material:

$$\frac{l}{2} = \frac{d}{2} = w = \sqrt[3]{\frac{V}{4}} \qquad (12)$$

The above formulas are mathematical ideals. In the majority of cases it will be found that the nature and dimensions of the articles packed will necessarily limit the application of these formulas. Assume 100 packages that are 2×2×3 in. in a corrugated cardboard carton.

Total volume V = 1,200 cu. in.

Hence, $w = \sqrt[3]{\frac{V}{4}} = \sqrt[3]{\frac{1200}{4}} = \sqrt[3]{300}$

w = 6.7 in.
$l = d = 2w$ = 13.4 in.

Mathematically, the optimum size for this volume is 13.4 in. long, 6.7 in. wide, and 13.4 in. deep. Using Equation (2)

Area = $A = 2\,wd+2ld+2wl+2w^2$
= 808.02 sq. in.

Actually, dimensions 13.4×6.7×13.4 in. will not contain 100 pieces 2×2×3 in. Packed five in a row in the 3 in. dimension and four across in the 2 in. dimension and five high in the other 2 in. dimension, dimensions will be 15×8×10 in. These dimensions become the three dimensions of one size of practical carton after adding a fraction of an inch to each dimension for clearance, 15⅛×8⅛×10⅛ in.

Again using Equation (2)

Area of board = $2\,wd+2\,ld+2\,wl+2\,w^2$
= 848 sq. in.

This is 848−808 = 40 sq. in. more material than the theoretical determined

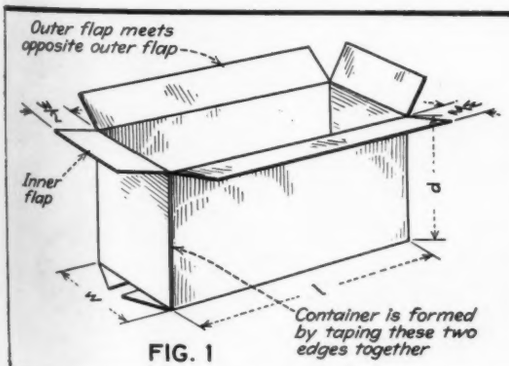


FIG. 1

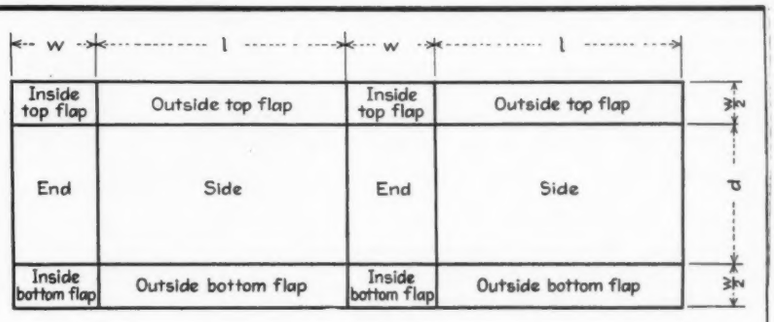


FIG. 2

Figs. 1 and 2—Standard container dimensioning. Left, method of joining; right, sheet surface required

above, or less than 5 percent over the mathematical minimum.

However, this is not the only size that could be used for 100 packages $2\times2\times3$ in., as they could be placed four along in the 3-in. dimension; five across in the 2-in. dimension; and five deep in the other 2-in. dimension. This stack is $12\times10\times10$ in. and after adding $\frac{1}{8}$ in. for clearance, size will be $12\frac{1}{8}\times10\frac{1}{8}\times10\frac{1}{8}$ in.

Once more applying Equation (2)

Area of board = 903 sq.in.

The area of material in the latter carton is 903—848 = 55 sq. in. more than in the former and 903—808 = 95 sq. in. more than the ideal size. Hence, the first practical size, namely, $15\frac{1}{8}\times8\frac{1}{8}\times10\frac{1}{8}$ in. is the most economical practical size of container.

The mathematical relation

$$w = \sqrt[3]{\frac{V}{4}}$$

is shown graphically in Fig. 3. Upon determining the required volume, it is a simple matter to locate the value of w.

Since in the theoretical case the relations between length, width, and depth are $l = d = 2w$, it is good practice to keep the length equal to the depth and both length and depth equal to twice the width. In all cases, the width should be the smallest dimension. The effect of this can best be illustrated if the container designed above is changed from $15\frac{1}{8}\times8\frac{1}{8}\times10\frac{1}{8}$ in. to $15\frac{1}{8}\times10\frac{1}{8}\times8\frac{1}{8}$ in. (length, width and depth respectively).

Applying Equation (2) to the latter carton,

area of board = 922 sq. in.

which is 922—840 = 82 sq. in. more than in the case $15-18\times8\frac{1}{8}\times10\frac{1}{8}$ in. Thus, it is evident that reductions in area can be easily affected simply by the proper rearrangement of dimensions.

The foregoing can be summarized in a few simple rules:

1. Determine total volume V
2. Substitute this volume in equation $w = \sqrt[3]{\frac{V}{4}}$ and determine w or determine w from Fig. 3. Let l and d each equal $2w$ if possible.
3. Width should be the smallest of the three dimensions.
4. Keep length l = depth d so far as possible. Where this is not possible, the length should be the longer of the two. Length and depth can be interchanged without altering the area of board involved but the length of the strip of tape used to close the container is the same as the depth of the carton, approximately.

Hence, in order to use the minimum amount of tape, the depth should be the shorter of the two larger dimensions.

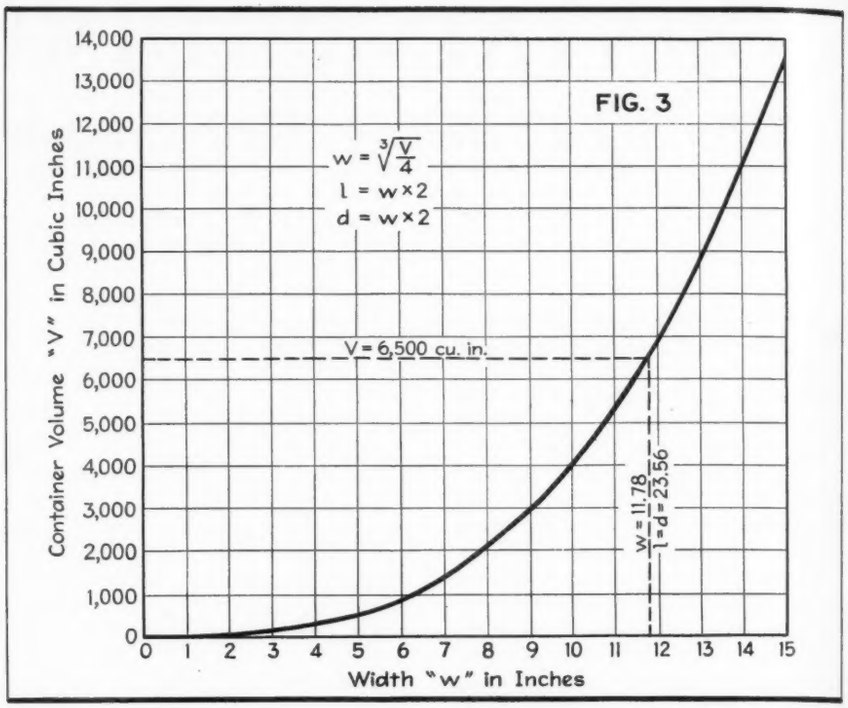
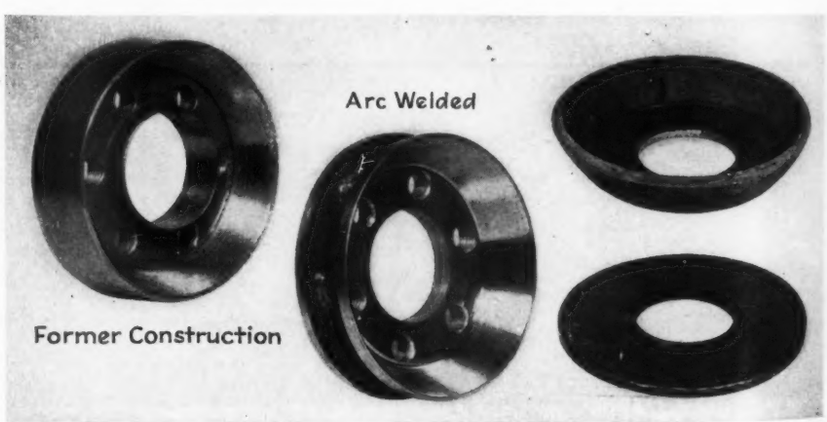


Fig. 3—Relation between container width and volume for optimum dimensioning

Welding Reduces Cost and Weight

ALTHOUGH REDESIGN of an entire machine for welded fabrication may not always be practical, savings in production cost may often be made through the use of arc welding for small component parts. The commutator cap illustrated below weighed 5 lb. 3 oz. in its former construction; it cost, including material and labor, $0.763. In the redesign, shown below right, the cupped part is welded to the flat disk with six intermittent arc welds spaced 60 deg. apart. Overall diameter is $5\frac{3}{8}$ in.; inside diameter is $2\frac{1}{4}$ in.; and overall thickness is 1 17/32 in. As redesigned, the part weighs 3 lb. 3 oz.; costs, including material and labor, are $0.320 per piece.

Weight saving amounted to two lb. or forty percent; cost saving was $0.443 per piece or fifty-eight percent. Another result of the redesign was an increase in operating efficiency since two lb. weight was removed from the armature load, resulting in a corresponding reduction in horsepower requirements. Data furnished by the Lincoln Electric Company.



Commutator cap former design at left; at right, welded design and component parts

LOG-LOG CHARTS

Simple Method for Their Construction

EMORY N. KEMLER
Purdue University

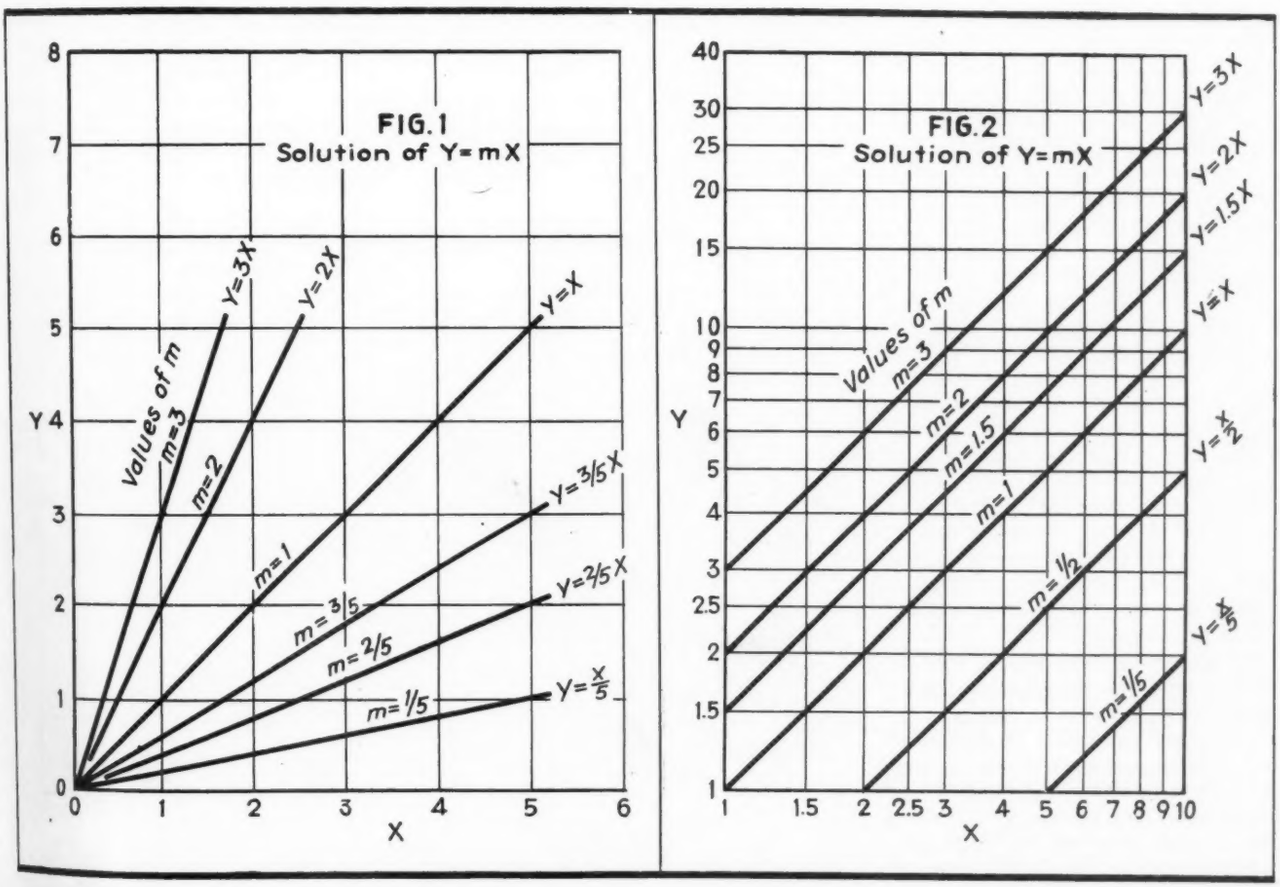
CERTAIN types of calculations can frequently be simplified by use of charts. The alignment chart, normally made up of parallel scales properly spaced and with proper scales, are frequently used. They also have many applications to problems where complicated functions are involved, but in these cases the simple parallel scales are no longer used.

An equivalent method which has not been subject to nearly as much study and which is frequently more easily applied is the method of setting up the variables on log-log paper. For simple functions this method has the advantages that it is easy to apply and is made on paper with a printed scale already on it and does not, therefore, normally require construction of two or more special scales.

Such charts are of greatest aid where repeated calculations are made or where a given calculation is to be made at infrequent intervals. The errors which normally occur in the arithmetical computation of long formulas are usually errors in units, omission of terms and slide rule errors. With a properly labeled chart the errors arising from incorrect units or omission of terms is greatly reduced. The accuracy will be about the same as that which can be obtained with a slide rule having an equal length for the log cycle.

The equation of a straight line on ordinary linear cross-section paper can be written as $y' = mx + b$. In this equation m is the slope of the line, and b is the value of y at which the line cuts the y axis, or where $x = 0$. If $b = 0$ the line goes through the origin, and the equation can be written as $y = mx$. Fig. 1 shows a plot of $y = mx$ for various values of m. This type of plot has a limited application since the range of values available on any scale is not very wide. Also the percentage error varies with the part of the chart being used. If, for example, values for y are desired when x is of the order of 10, for $m = 3$, the chart in Fig. 1 would have to be extended 6 times its length. Also for very small values of x, y will be hard to read accurately.

If we take the log of each side of the equation $y = mx$ we get $\log y = \log m + \log x$. This resulting equation will be equivalent to the original equation of a straight line having a unit slope, that is, being at an angle of 45 deg. with the x axis. If there we plot $\log y$ and $\log x$ as coordinates, the resulting family of curves for varying values of m will be as is shown in Fig. 2. On this chart values can be read with a constant



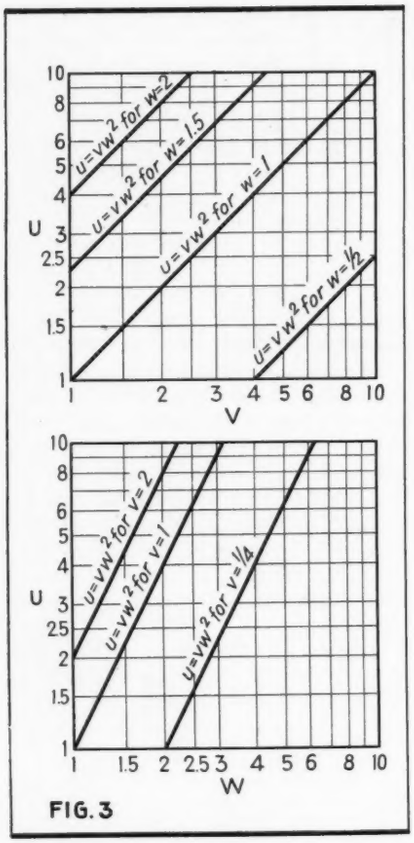


FIG. 3

percentage error for any or all values. Furthermore the two charts can be drawn for any given range with equal ease. The range may be for 1 to 10, 1,000 to 10,000 or 0.0001 to 0.001, on either scale, or a correspondingly larger range if multi-cycle paper is used.

For equations of the form $U = VW^n$ we get upon taking logs, the equation $\log U = \log V + n \log W$. This equation is identical in form with the general equation for a straight line if we let $\log U = y$, $\log V = b$, etc. In this case if W corresponds to the x term the slope of the straight line will be two. If $\log V$ is set equivalent to x in the straight line equations, the lines will have a slope of unity. In this case the W lines will be spaced n times the units apart. Fig. 3 shows in the lower section typical lines for $U = VW^2$ with W taken corresponding to x. The upper section shows V taken corresponding to x and the W terms spaced n times as far apart as normal. Fig. 3 has been drawn for $n = 2$ as indicated.

The use of the above general method can be extended to the solution of more complicated equations or to equations involving more variables. In the case of several functions the final answer can be obtained by first multiplying two of the variables then combining this result with a third, etc., until the final solution has been obtained. The order of selection of variables normally makes little difference. In some cases, however, a more compact chart can be drawn for one order of variables, than when using another order.

To illustrate the application of these charts the calculation of the bending moment which a rectangular beam of width b, and depth d can stand at various working stresses will be solved. This equation can be written as

$$M = (1/6)\ (Sbd^2)$$

The method of attack is normally to select any two variables and find their product. If for simplicity we choose S and b we have $Z = Sb$. A chart for calculating Sb will be similar to that shown in Fig. 4. If we wish to include the 1/6 term we find

$$U = (1/6)\ (Sb)$$

the chart will be exactly the same except that the lines have been shifted on Fig. 5 as indicated.

Once the product $1/6\ Sb$ has been calculated the product of Ud can be worked on a chart. This can best be done as is indicated in Fig. 6. If we trace from the value $U = 1/6\ Sb$, which can in effect be read on the left-hand scale as in Fig. 5, to the value of d, the value of $1/6\ Sbd^2$ can be read on the lower scale as indicated. In this case the values of d read down rather than up. The reason for this being that the x and y coordinates are in effect changed. The lines are spaced as in the top part of the Fig. 3 chart.

While rules could be given to determine the slope and spacing of lines, it is actually much easier to work out the chart a little at a time, to determine the location and order of lines. Some experimenting is frequently desirable to determine the proper order of lines to give the most compact chart. As mentioned above, skill in use of this method can best be obtained by use of the method and by study of other charts of this nature. Several of the log-log type charts have been given in the PRODUCT ENGINEERING data book examples. By reference to them and study of this discussion, general ideas on the construction of such a chart can be developed for many types of formulas.

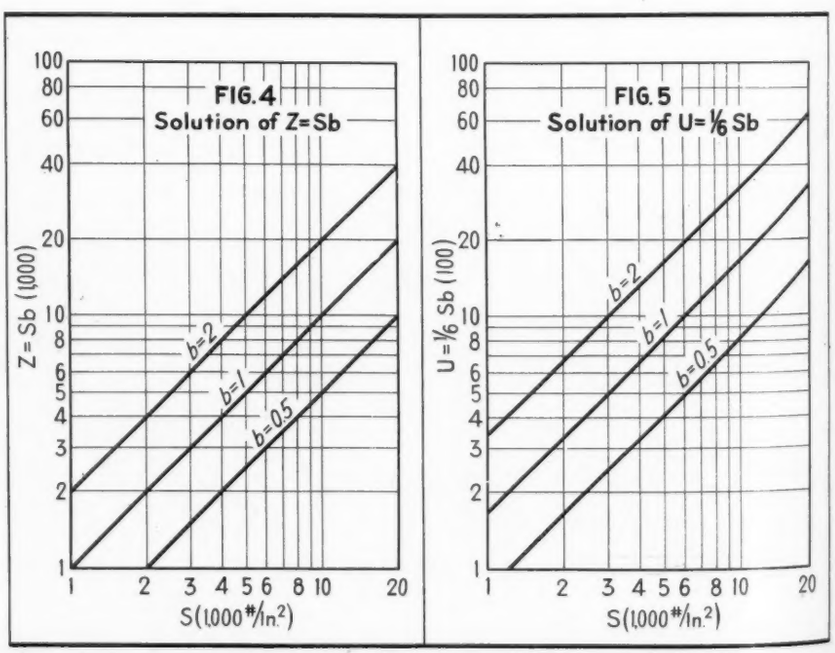
To illustrate a more complicated type of chart the general solution and position of lines for a formula involving many variables is indicated in Fig. 7. The formula used is that for indicated horsepower,

$$IHP = \frac{a\ s\ L\ A\ N}{l \times 12 \times 33{,}000}$$

where

a = area of indicator card in inches²
l = length of indicator card in inches
s = indicator spring scale
L = length of stroke in inches
A = area of piston in square inches
N = number of strokes per minute

This chart indicated in Fig. 7 considers the variables in the order a, l, s, L, A, N. The constant term is introduced in the first set or in the a, l part of the chart. Actually these constants can be introduced at any place in the series of charts which are in effect shown. Frequently the chart can be made more compact by a judicious choice of the place of introduction of the constant term, in whole or in part. In Fig. 7 has been shown the various quantities which could be read off the chart desired.



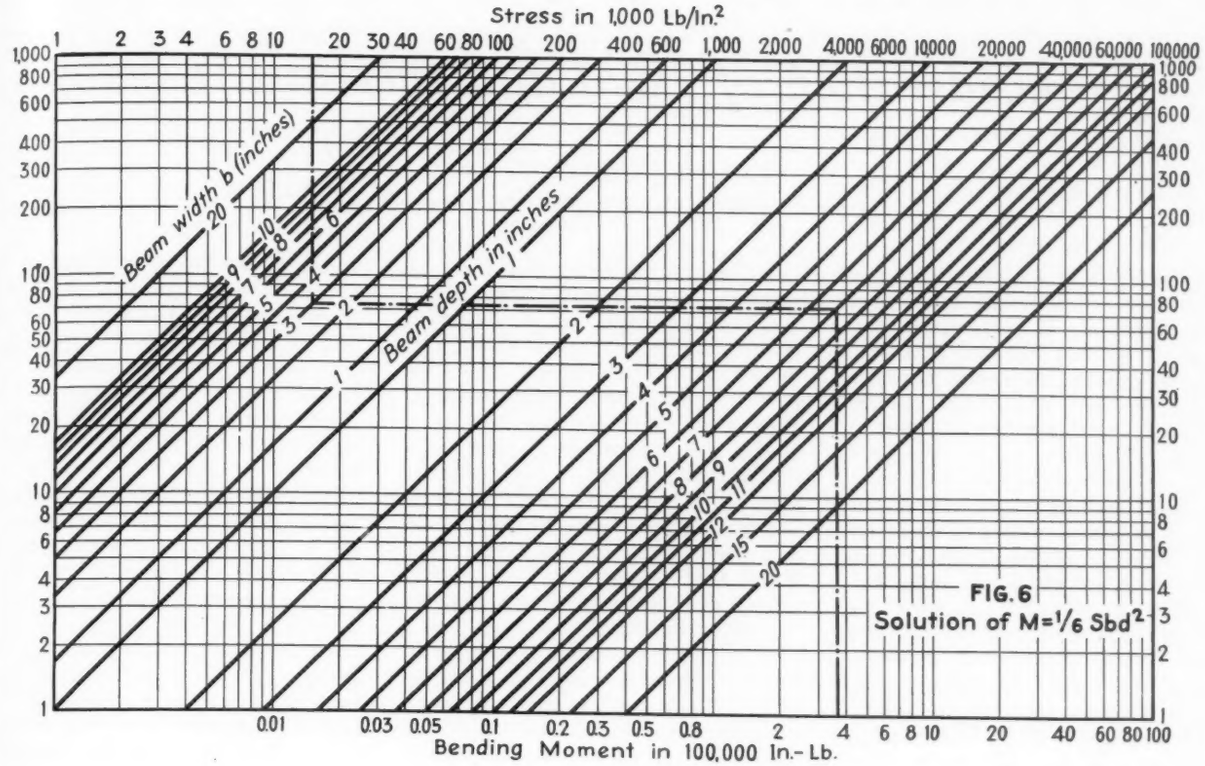


FIG. 6
Solution of $M = 1/6\ Sbd^2$

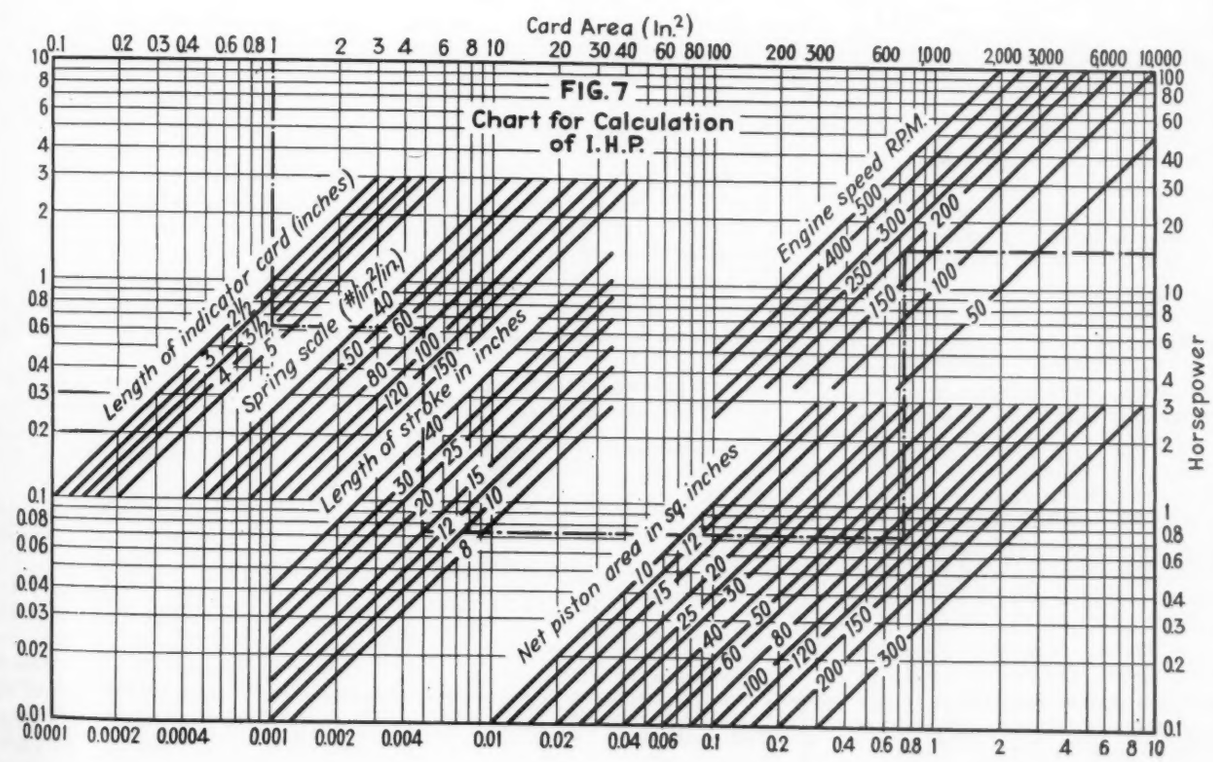


FIG. 7
Chart for Calculation of I.H.P.

SYNCHRONOUS MOTORS

Consideration of Load and Inertia Effects

F. K. BRAINARD

A. C. Engineering Department, Allis-Chalmers Manufacturing Company

IN the larger sizes of alternating current motors the synchronous type has a number of advantageous features. Its starting characteristics can be made suitable for nearly any constant-speed drive; when running it can improve a low plant power factor; it has high efficiency; and for low-speed drives its price is comparatively low. But when selecting a synchronous motor care must be exercised to select one with starting as well as running characteristics suited to the particular machine to be driven. Conversely, the necessary starting characteristics of a motor are determined by the power required by the driven machine during the starting period and by the inertia (flywheel effect or WR^2) of the rotating parts. As shown later, the selection of suitable characteristics of a proposed synchronous motor may depend to a considerable extent upon the inertia of the load. From the standpoint of starting it is always desirable to reduce the inertia as much as possible, but sometimes it is necessary to increase it to obtain satisfactory operation under load.

A synchronous motor is quite similar in construction to a synchronous a.c. generator. Alternating current power is delivered to the armature and mechanical power is taken from the shaft, instead of vice versa, as is the case with a generator. Direct current excitation is supplied to the field winding in either case. Either the field or the armature may be the revolving element, but like the generator and for the same reasons it is usually built with the field revolving.

Before a synchronous motor can operate normally it must be brought into synchronism with the supply circuit, connected to it, and fully excited. The usual method of doing this is to equip the pole faces with an amortisseur winding, which is similar to the squirrel cage winding of a squirrel cage induction motor, and to start by connecting the armature to the a.c. power supply, the field circuit being closed through a suitable resistance. The alternating current in the armature produces a magnetic field which revolves around the perphiphery of the armature at the rated speed of the motor. This revolving magnetic field induces currents in the amortisseur winding and in the closed field circuit, the reaction of which upon the magnetic field produces the torque of the motor during the starting period. Then after the motor has reached a certain speed, generally about 95 percent of full speed, direct current excitation is applied to the field winding, and the motor will go to full speed and will carry loads up to the pull-out torque of the motor.

Obviously, the torque developed by the motor during the starting period must be enough to bring it sufficiently close to synchronism to enable it to pull into step when direct current excitation is applied; and this will depend upon the power required by the driven machine at this point, and upon the other factors mentioned later in this article.

Fig. 1 shows the speed-torque curve of a certain synchronous motor and the centrifugal pump which it drives. The pump cannot be unloaded for starting. As is seen, the torque developed by the motor greatly exceeds that required by the pump until 97 percent speed is approached, at which point the torques are equal. Direct current excitation is applied at about this speed and the motor pulls into step readily.

The most desirable shape of speed-torque curve depends upon the requirements of the driven machine. A high-resistance amortisseur winding results in relatively high starting and low-pull-in torques, while a low-resistance amortisseur winding results in low starting and high pull-in torques. Fig. 2 shows typical speed-torque curves of motors suitable for various kinds of service.

Because of its effect upon the power system and upon the size and cost of the motor, it is desirable to reduce the load as much as possible during the starting period. Frequently it can be materially reduced quite easily as is the case with most reciprocating compressors, positive pressure blowers, and centrifugal pumps or blowers. Some machines, however, such as cement mills cannot be unloaded for starting. Others, such as mine fans, which have large rotating parts can be partially unloaded. Still others, such as induced draft fans having heavy rotating parts cannot be unloaded very much. Each of these cases requires careful consideration, particularly those where both large starting loads and heavy rotating parts are involved. If the starting and pull-in torque requirements are severe it may be desirable to use a mechanical, magnetic, or hydraulic clutch of some kind between a standard synchronous motor and the load. This is sometimes less costly and more satisfactory than a special motor suitable for use without the clutch. In this case the motor will be started light, brought up to speed, and fully excited before engaging the clutch. Care must be exercised to see that the motor has enough pull-out torque to prevent its being pulled out of synchronism when the clutch is engaged.

Excessive flywheel effect manifests itself in two distinct ways, both of which make starting more difficult.

The first is to increase the total energy loss in the amortisseur winding during the starting period and consequently the heating. This is because the acceleration is lower and the high amortisseur loss persists for a longer period. In an extreme case both armature winding and amortisseur winding might be damaged.

The second effect is the reduction of

Synchronous Motor Applications

Air compressors
Ammonia compressors
Ball mills
Blowers, positive pressure
Blowers, centrifugal
Cement mills
Dryers, cement mills
Fans, induced draft
Fans, mine
Frequency changers
Lineshafts
Motor-generator sets
Pumps, centrifugal
Pumps, reciprocating
Rock crushers
Rubber mills
Steel mill drives
Ship drives

the pull-in torque of the motor. After reaching the highest speed at which it will operate as an induction motor, the application of d.c. excitation introduces an additional component of torque which alternates at slip frequency. This torque causes pulsations in speed above and below the mean, magnitude of which depends upon flywheel effect, other things being equal. If the flywheel effect is increased the pulsation in speed will be less, and the initial speed at the time that excitation is applied must be increased if the motor is to pull into step, because these pulsations in speed are the means by which the gap between induction and synchronous operation is bridged. The curves in Fig. 3 explain this effect.

The maximum slip from which it is possible to pull in depends upon the design of the motor, the inertia (flywheel effect) of the rotating parts, the characteristics of the system from which power is drawn, and the phase angle at which voltage is applied. Shoults, Crary, and Lauder (*A.I.E.E. Transactions*, Vol. 54, 1935, p. 1,385) give the following expression for average slip for the most unfavorable phase angle:

$$s < k_2 \sqrt{\frac{E_d\, e}{\pi f H (X_d + x_1)}} \qquad (1)$$

where

s = average per unit slip before excitation is applied,
k_2 = 0.55 to 0.60
E_d = per unit direct axis voltage corresponding to field excitation
e = system per unit voltage back of line reactance x_1
x_1 = per unit line reactance to point of constant voltage
f = line frequency in cycles per sec.
H = inertia constant

$$= \frac{0.231 \times WR^2 \times N^2}{KVA \times 10^6}$$

X_d = per unit direct axis synchronous reactance
N = revolutions per min

As shown by Equation (1), the maximum permissible slip at the time excitation is applied varies inversely as the square root of the flywheel effect. This means that if the flywheel effect is large, it becomes necessary to bring a given motor closer to synchronism than would otherwise be necessary; and the load must be correspondingly reduced.

In order to obtain a basis for the comparison of the relative flywheel effects of various loads at various speeds, the N.E.M.A. has adopted values of "normal load WR^2" based upon the empirical expression:

$$\text{normal load } WR^2 = 3.75 \times \frac{10^5\,(hp)^{1.15}}{N^2} \qquad (2)$$

This expression represents a value of inertia of driven load that can generally be pulled into step without difficulty if the torque load is not greater than the "nominal pull-in torque" as defined below.

Pull-in torque is the maximum constant torque under which the motor will pull its connected inertia load into synchronism. Equation (1) shows that this depends upon the flywheel effect of the load and the characteristics of the power system, as well as upon the design of the motor itself, and cannot be determined unless these quantities are known. However, it has been found in nearly all cases that 5 percent slip is permissible; that is, if the motor is designed so that it will reach 95 percent speed as an induction motor when

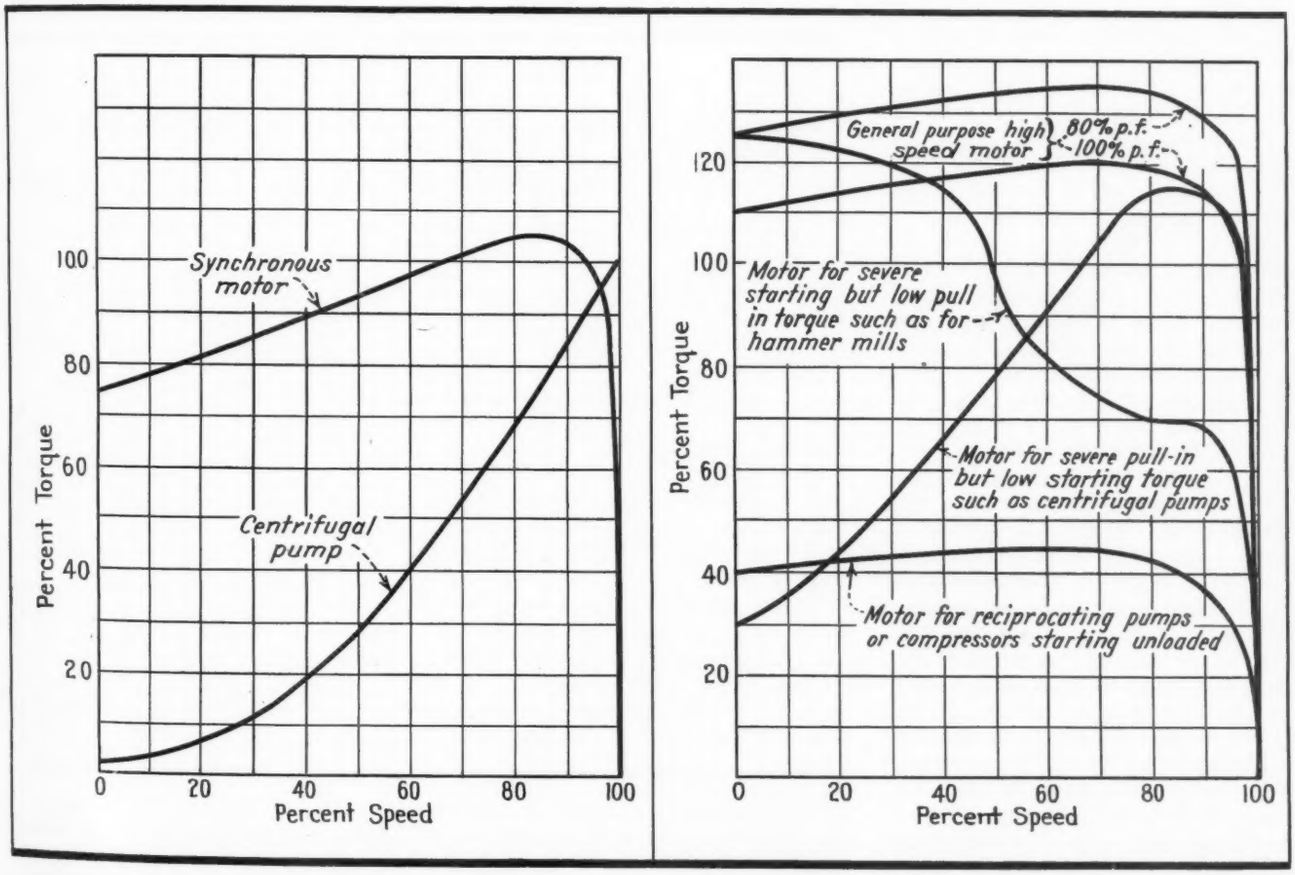


Fig. 1—Starting as an induction motor under the influence of an amortisseur winding and the closed field circuit, the speed of a synchronous motor builds up to about 95 percent of synchronous speed. At this point d.c. excitation is applied to the field and the rotor pulls into step with the revolving magnetic field produced by the field current and reaches full synchronous speed, at which it continues to run at any load up to the maximum pull-out torque of the motor which is usually 150 percent or more. Fig. 2—Typical speed-torque curves of motors suitable for various kinds of service

developing the specified torque, it will usually pull in when excitation is applied. For this reason "nominal pull-in torque" has been defined by N.E.M.A. as the torque at 95 percent speed when operating as an induction motor without excitation.

By assuming an adequate power system and by making certain other approximations, the ratio of "nominal pull-in torque" to "pull-in torque" can be expressed as a function of the horsepower and power factor ratings of a motor and the ratio of the WR^2 of the load to the "normal load WR^2." This is shown by the curves in Fig. 4, and is useful for approximate work. If accurate results are desired, however, Equation (1) should be used.

As an example of the use of the above curves, let us assume that it is desired to drive a certain induced draft fan with a 300 hp., 80 percent power factor, 1,200 r.p.m. synchronous motor. During normal operation with the exhaust gases hot it requires 275 hp. to drive the fan. Dampers are provided to reduce the load during starting, but because of the necessity of starting cold the load cannot be reduced to less than 225 hp. at the close of the starting period. The flywheel effect of the fan is 9,000 lb.-ft.^{2} which is 49 times the normal load WR^2 for the 300 hp. motor, as given in tables of normal WR^2 for synchronous motors. Values for A and B are 2.37 and 0.90, respectively, from Fig. 3. The required nominal pull-in torque will then be approximately:

$$225 \text{ hp.} \times 2.37 \times 0.90 = 480 \text{ hp.}$$

Therefore, a 300 hp., 80 percent power factor motor having a nominal pull-in torque of about 160 percent instead of the N.E.M.A. standard 125 percent pull-in torque must be used. This illustrates how important it is to give careful consideration to the flywheel effect of the load in applying synchronous motors.

So far we have been considering only the detrimental effects of the inertia of the revolving parts, but in many cases it is necessary to increase the inertia to obtain satisfactory operation of the synchronous motor-driven unit. This is true in spite of the fact that the synchronous motor is essentially a constant-speed machine.

Additional flywheel effect is frequently used to advantage to reduce the variations of power and current drawn from the line which would otherwise result. If the load is a periodically pulsating one, such as that of a reciprocating pump or compressor, these variations would frequently be serious if suitable values of flywheel effect were not used. Excessive variations of current result in corresponding variations in voltage which may cause objectionable flickering of lamps supplied from the same circuit. In extreme cases damage to the motor is possible. The American Standards Association rule covering this is as follows:

"When the driven load such as that of reciprocating type pumps, compressors, and others require a variable torque during each revolution, the combined installation shall have sufficient inertia in its rotating parts to limit the variations in motor armature current to a value not exceeding 66 percent of full load current.

"The basis of determining the variation shall be by oscillograph and not by ammeter readings. A line shall be drawn on the oscillogram through the consecutive peaks of the current wave. The variation is the difference between the maximum and minimum ordinates of this envelope. This variation shall not exceed 66 percent of the maximum value of the full load current of the motor. (The maximum value of the motor armature current to be assumed as 1.41 times the rated full load current.)"

In many cases the percentage variation in power and current drawn from the line is less than the percentage variation in torque causing it. This is not always the case, because there may be partial resonance between the frequency of certain impulses and the natural frequency of the unit on the power system.

Driving a compressor and its flywheel from a large power system by means of a synchronous motor is analogous to driving it from the same prime mover through gearing and a somewhat flexible line shaft. It runs at an average speed proportional to the speed of the prime mover, but there may be momentary variations in speed above and below the average because of the flexible connection. If the prime mover is large compared with the compressor, it is easy to see that the line shaft and flywheel in the mechanical analogy constitute a torsional pendulum which will oscillate at a definite frequency determined by the moment of inertia of revolving parts and the stiffness of the line shaft. If the compressor produces impulses the frequency of which is close to this natural frequency, partial resonance occurs and violent oscillations may result. In this case the variations in torque of the line shaft will be

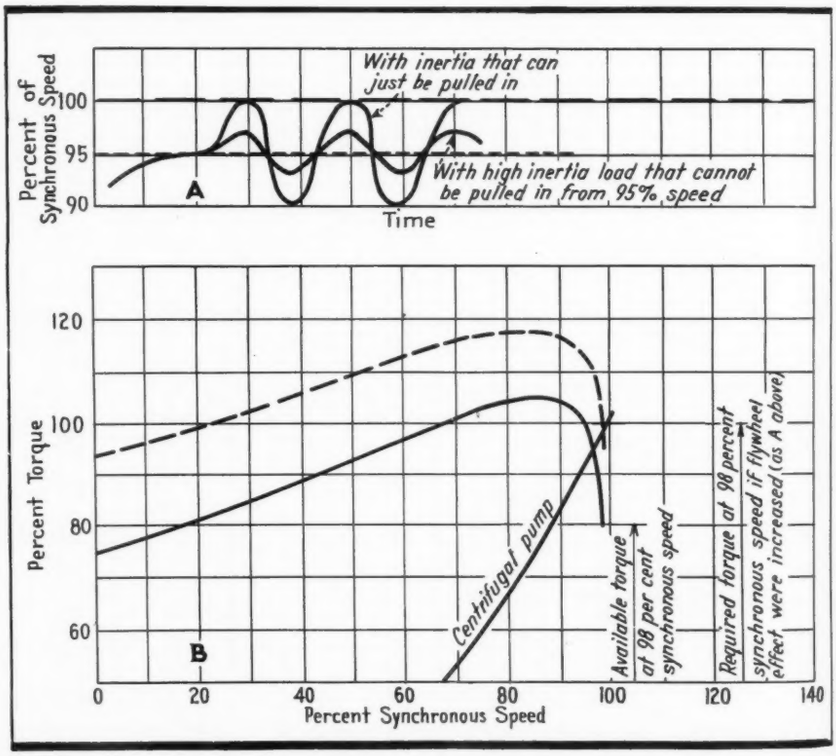


Fig. 3—Diagram A illustrates approximately the effect of the inertia of the load on the ability of the motor to pull into step. When d.c. excitation is applied to the rotor, pulsations are set up as shown. If the inertia is within proper limits the motor will pull into step and run at 100 percent synchronous speed when the proper phase angle is reached. If the inertia is too great the pulsations will never reach 100 percent speed. To do so it would be necessary in this example to run the motor on the starting winding up to 98 percent synchronous speed before applying d.c. excitation. If applied to the pump in Fig. 1 it would have the effect shown at B. The motor design would have to be changed to get approximate characteristics shown on the dotted line

much greater than the variations in torque of the compressor. If these oscillations are very great, the line shaft in the mechanical analogy might break, which corresponds to the synchronous motor dropping out of step. The torque per electrical radian, P_r, is analogous to the stiffness of the shaft.

The A.S.A. formula for natural frequency of a synchronous machine on a large system is:

$$F = \frac{35{,}200}{N}\sqrt{\frac{P_r \times f}{WR^2}} \qquad (3)$$

where

P_r = synchronizing power in kilowatts per electrical radian as defined in A.S.A. rules
f = frequency of the system in cycles per sec.
N = rated speed in r.p.m.
WR^2 = flywheel effect of unit in lb. ft.^{2}

A factor has been adopted by the N.E.M.A. for use in determining the flywheel effect needed for various compressors. Designated by C in Equation (4), it provides a means for comparing the relative values of flywheel effect of units of various sizes and speeds

$$C = \frac{0.746 \times WR^2 \times N^4}{P_r \times f \times 10^8} \qquad (4)$$

Values of C corresponding to 66 percent power pulsation have been adopted by N.E.M.A. and A.S.A. for various types of compressors having various methods of unloading (A.S.A. Standards C-50-1936). By solving Equation (4) for WR^2 the following convenient expression is obtained:

$$WR^2 = \frac{1.34\ P_r f\ C}{(N/100)^4} \qquad (5)$$

By using values of C from the A.S.A. rules in Equation (5), the limiting values of WR^2 are readily obtained. From this it is seen that the flywheel effect necessary for any compressor driven by a synchronous motor drawing power from a large system varies directly as the compressor factor C, the frequency of the system, and P_r; and inversely as the fourth power of the speed. Since P_r varies almost directly with the rated output of the motor, the WR^2 will also, other things being equal.

The permissible values of C for a given compressor depend almost entirely upon the shape of the tangential effort curve. This can be shown as follows:

Solve Equation (3) for WR^2 and substitute this in Equation (4). We then obtain:

$$\frac{N}{F} = \sqrt{\frac{C}{9.25}} \qquad (6)$$

The ratio of speed to natural frequency thus depends upon C, and since the ratio of electrical power variation to torque variation for each harmonic of the torque curve depends principally upon the ratio of speed to natural frequency

Standard Torque Characteristics of Synchronous Motors

Motor Size	Poles	Torque (Percent of rated full-load torque)		
		Starting	Pull-in	Pull-out
General Purpose				
Up to 200 hp (1.0 power factor)	4	110	110	150
	6–14	110	110	175
Up to 150 hp. (0.8 power factor)	4	125	125	200
	6–14	125	125	250
Large High Speed				
250–500 hp. (1.0 power factor)	4–14	110	110	150
250–500 hp. (0.8 power factor)	4–14	125	125	200
600 hp. and larger (1.0 power factor)	4–14	85	85	150
600 hp. and larger (0.8 power factor)	4–14	100	100	200

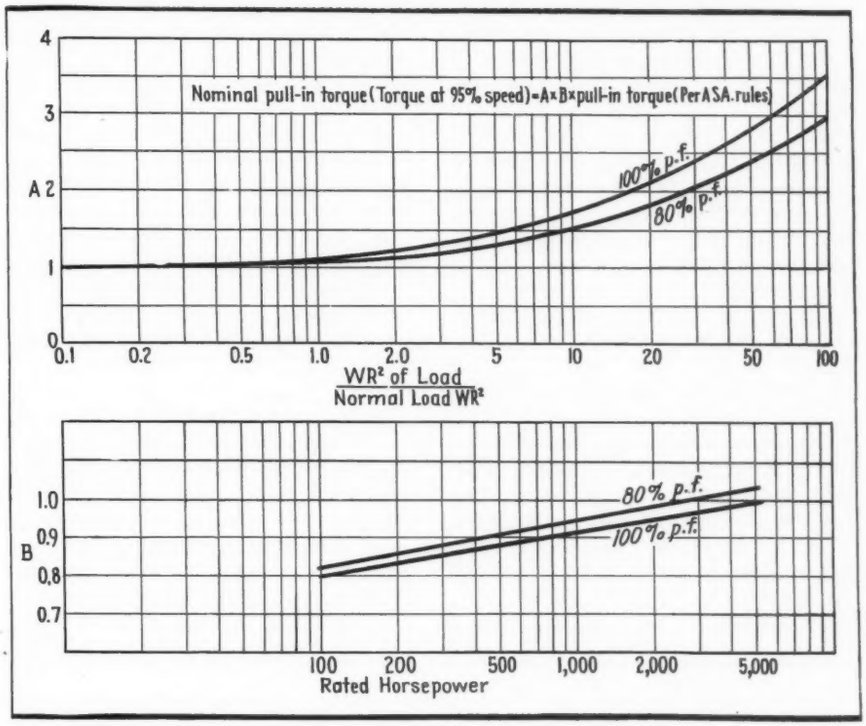

Nominal pull-in torque (Torque at 95% speed) = A × B × pull-in torque (Per A.S.A. rules)
A
B
100% p.f.
80% p.f.
WR² of Load / Normal Load WR²
Rated Horsepower

Fig. 4—These curves are useful for determination of the ratio of "nominal pull-in torque" to "pull-in torque" of a motor when WR^2 of driven machine is known

it will depend upon C also. Frequently a judicious modification of the design of the compressor will make a material difference in the permissible values of C, and for this reason close cooperation between the compressor and synchronous motor designers is often advantageous and desirable.

As an illustration of the frequencies to be avoided, consider a horizontal duplex double-acting ammonia compressor with cranks at 90 degrees. This compressor will have an appreciable unbalanced impulse occurring once per revolution, resulting mainly from the angularity of the connecting rods; there will be no impulse occurring twice per revolution since whatever is produced by one cylinder will be neutralized by the other if the two cylinders are operating exactly alike; there will be a small impulse three times per revolution; and there will be a large impulse four times per revolution resulting from the four compression periods. Thus the impulses with which resonance must be avoided by a wide margin are those corresponding to the speed of the unit and to four times the speed of the unit. Three times the speed of the unit should also be avoided but not necessarily by as wide a margin. From Equation (5) these correspond to the following values of C to be avoided: 9.25, 1.03, 0.58. The A.S.A. standard for this application is C = 2.0 to 6.0 or 12.0 and over, which meets the requirements mentioned above.

Question and Comment

Elastic Limit of Flat Spring Material

A. THOMPSON

In designing flat springs it is necessary to assume certain definite values for the modulus of elasticity and for the elastic limit. Modulus of elasticity is a fairly constant property of each material but the elastic limit varies greatly, depending on the degree of cold working or the temper of the metal.

If a sufficiently long sample strip of the material is available, the method described below permits an experimental determination of elastic limit without the use of a testing machine or any instruments beyond a scale and micrometer.

First let us consider a strip of spring material clamped at one end and loaded lengthwise at the free end (Fig. 1). As the load P is increased sufficiently the strip will buckle first to position (2), and then to position (3), bringing the free end on the same horizontal line with the point of clamping A. In this position the angle α will be equal to 130 deg. 43 min., and $y_a = 0.7833\ l$ as follows from the theory of a buckled bar (S. Timoshenko, Theory of Elastic Stability, 1936, p. 72).

If now we take a strip of length $L = 2\ l$ and holding it by the ends bend it until the ends just touch, we then will have exactly the same conditions as for the buckled bar of length l described above. It can be shown that the bending stress at point A

$$S = 4.2192\,\frac{Et}{L}$$

Thus this stress depends only on the modulus of elasticity E, thickness of the strip t and its total length L.

If now we have to determine the elastic limit of material, we can proceed as follows: we first estimate the elastic limit, assuming it somewhat lower than expected, assume E and measure t. Then

$$L = 4.2192\,\frac{Et}{S}$$

and we cut off a piece of this length. We then bend it, bringing the ends together, and releasing find that there is no permanent deformation. Next we bend it again but this time let one end slide over the other, as shown in Fig. 2, thus reducing the active length and measure the length ΔL which has to be deducted from L. We keep reducing the active length until a permanent set is observed, and we know then the corresponding length L and can figure out the stress at the elastic limit. Knowing the elastic limit, the allowable stress can be estimated with a good degree of certainty.

It should not be expected of course, that this result will be as accurate as one obtained by the use of a testing machine. Nevertheless it will be a great deal more accurate for the material in question than a figure from a handbook or catalog, where the elastic limit of cold worked material, if at all given, is usually shown "from-to" within wide limits.

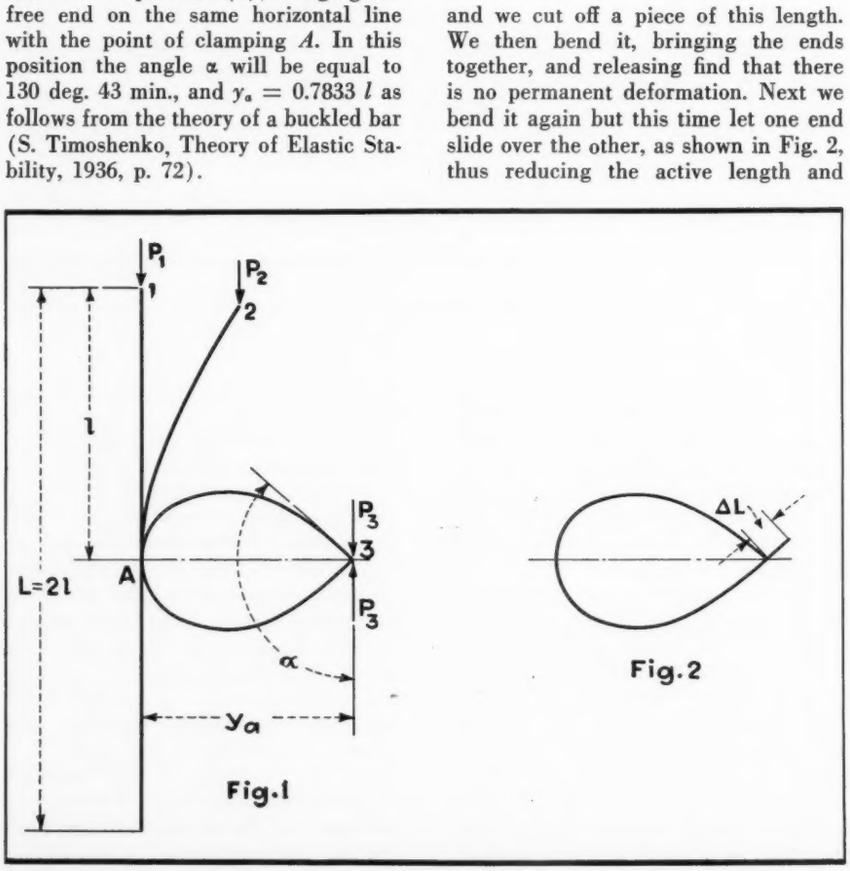


Fig. 1

Fig. 2

Discusses Properties Of Polyvinyl Alcohol

To the Editor:

In your issue of March, 1941, on page 151, appeared a Reference Book Sheet entitled, "Properties of Synthetic Rubber-Like Materials," signed by J. Delmonte. As manufacturers of Resistoflex PVA material (polyvinyl alcohol compound), we were keenly interested in this tabulation. In the references to polyvinyl alcohol we noticed certain discrepancies which we presume you will welcome having called to your attention.

Under "Chemical," the statement is made that polyvinyl alcohol is "resistant" to gasoline, oils, greases, etc. The word "resistance" is a relative term and has been badly abused by application to materials which swell as much as 40 or 50 percent and in some cases even 100 percent, from contact with various chemicals. Resistoflex PVA material is almost entirely unaffected by any of the solvents listed.

The statement is also made under this section of the tabulation as well as under the one entitled, "Limitations," that polyvinyl alcohol is not resistant to mixtures involving any water, glycerine or glycerol, or alcohol. As a matter of fact, Resistoflex PVA is being used extensively today for fuel lines and oil lines on aircraft, automotive, diesel engines and industrial equipment, where quite a substantial amount of water is present either in the form of

condensate or crankcase dilution. No harmful effects have ever been noticed from the small amount of water usually present in such commercial applications.

As a matter of policy, the material is not offered for uses in connection with compressed air, even where a filter is used, or for conducting acids, alkalies, or other aqueous solutions. In this same connection, Resistoflex PVA is highly resistant to both methanol or ethanol in their anhydrous form, although both they and other water-soluble organic solvents have a tendency to leach plasticizers out of the material to some extent.

This information is being submitted to you in no critical sense but merely in the interest of accuracy.

—EDGAR S. PEIERLS, *President*
Resistoflex Corporation

Total Load on Bolts

WALTER M. HAESSLER
American Car & Foundry Company

In the "Question and Comment" column in the February 1941 number, I came across the inquiry made by R. D. Thomsen of Milwaukee, Wis., concerning the total load on bolts and screw fastenings in general. I have wrestled often with this problem and I would like herewith to submit a very simple analysis of this perplexing problem.

Suppose a plate suspended by two bolts has a load $2\ W$ hung midway between the bolts (Fig. 1). Then each bolt is subjected to a simple axial tension $T = W$ and the case presents no ambiguity. Commonly, however, bolts are used to connect pieces to some extent elastic which are bolted or screwed tightly together with consequent compressibility, hence creating an initial tension in each bolt. This tension is balanced by the elastic reaction of the pieces fastened together. If C_i is the initial thrust per bolt of the compressible rubber block (Fig. 2) then C_i must also be the initial tension in each bolt due to screwing up. If the extension of the bolts is α and the compression of the rubber block is β per unit of load then the initial compression of the rubber block due to screwing up is βC_i and the corresponding extension of the bolts is $\alpha\ C_i$.

Now, after screwing up, a load $2\ W$ is symmetrically applied (Fig. 3). In general, the tension of the bolts will increase to a value T between (W) and $(W + C_i)$ and the additional extension of the bolts will *diminish* the compression of the rubber block. If $2\ W$ is increased till the additional extension of the bolts is equal to or greater than the initial compression of the rubber block then the case reverts back to that represented in Fig. 1. In most practical cases the bolts must be screwed up initially so as to prevent this latter condition.

When, as in Fig. 3, the load $2\ W$ is applied, the additional extension of the bolts is $\alpha\ (T - C_i)$ and the compression of the rubber block diminishes to $\beta\ C_i - \alpha\ T - C_i)$. If C is the thrust of the rubber block per bolt in this latter condition then

$$\beta C = \beta C_i - \alpha\ (T - C_i)$$

$$C = C_i - \frac{\alpha}{\beta}\ (T - C_i)$$

$$T = W + C = W + C_i - \frac{\alpha}{\beta}\ (T - C_i)$$

$$T = \left(\frac{\beta}{\alpha + \beta}\right) W + C_i \ldots\ldots (1)$$

Many interesting deductions can be made from equation (1)

(a) If $\beta = 0$ or in other words if the materials being clamped were infinitely rigid then $T = C_i$; i.e., the bolt tension would never exceed the initial screwing of load.

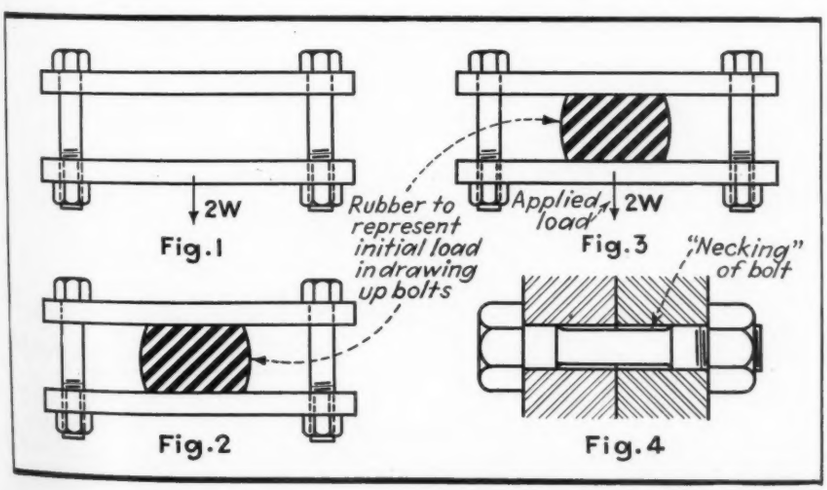
(b) If $\alpha = 0$ or in other words if the bolt was infinitely rigid then $T = W + C_i$ and the consequent bolt tension would be greater than in (a) by an amount $= W$.

(c) In all practical actual cases neither α nor β are zero but their relative magnitude is of great importance and the relationship $\frac{\beta}{\alpha+\beta}$ is the cause of the importance.

(1) If the bolt is considerably elastic in relation to the materials being clamped, i.e. α (high) and β (low) then the total load will only be slightly more than the initial screwing-up load. Fig. 4 illustrates this condition. The action of "necking" the bolt will result in increasing α in relation to β and despite the lower cross-sectional area the bolt will be less highly stressed than if the original shank of the bolt had been left unchanged. Examples of this occur in connecting rod "big end" bolts and other numerous cases. Likewise the same result can be attained by using long bolts or studs to get flexibility such as in cylinder head studs. A striking example of this was the case of a pneumatic jackhammer manufacturer who found his cylinder head bolts were breaking frequently. His original design had short bolts holding each cylinder head to each end of the flanged cylinder. By substituting long through-bolts which simultaneously fastened both cylinder heads to the flanged cylinder he found that his troubles were over.

This problem can be carried on to include many examples but time and space would not permit.

I would like to start further comment on this subject.

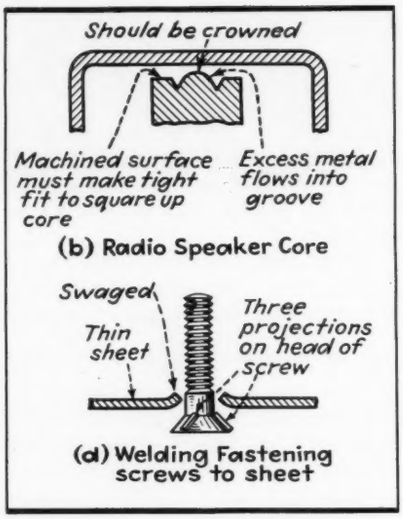


Correct Designs For Projection Welding

To the Editor:

Projection welding is an excellent method for fastening parts, but two of the sketches shown in the lower left-hand corner of page 203 of your April "Designs for Speedier Production" number probably wouldn't give good results if applied exactly as shown. Actually, the drawings should have been made as sketched below.

A projection weld such as is shown in (b) should concentrate the welding current through an initial point contact between the two parts. This generates


Should be crowned
Machined surface must make tight fit to square up core
Excess metal flows into groove
(b) Radio Speaker Core
Swaged
Thin sheet
Three projections on head of screw
(d) Welding Fastening screws to sheet

sufficient heat to start a good weld, which then spreads over a larger area until a weld of the desired strength is achieved. To obtain this point contact on the radio speaker core and on similar parts, the end of the core should be rounded off as shown here. Common round-headed bolts, as shown in sketch (d) in April, page 203, are "naturals" in this respect.

In projection welding fastening screws to thin sheets as illustrated in sketch (d), page 203, the sheet should be swaged as shown here.

In projection welding the bimetallic contacts shown on page 202, it is necessary first to form slight projections on the backing metal of the contacts. Shape and number of these projections depends upon the contact's size and current-carrying capacity. Bimetallic silver contacts with monel or steel backs have successfully been welded to phosphor-bronze springs, brass, beryllium-copper, steel and monel. When welding copper wire to lugs or to another wire, use tungsten-faced electrodes on the copper wire; bare copper electrodes conduct the heat away too rapidly.

—R. T. Gillette
General Electric Co.

[*Editor's Note*—Thanks, Mr. Gillette. Thanks, too, for all the extra information on projection welding of typical small electrical parts.]

Better Late Than Never

To the Editor:

The Bureau of Aeronautics' copy of the March issue of Product Engineering has just reached me and I note that it contains my article on "Aircraft Auxiliary Power." However, the statement to the effect that "the opinions or assertions contained therein are the private ones of the writer and are not to be construed as official or reflecting the views of the Navy Department, or the naval service at large" (Navy Regulations, Article 113), has been inadvertently omitted. Please publish a note to this effect in an early issue.

—Vernon H. Grant
Navy Department, Bureau of Aeronautics

Describes Another Method To Reduce Lamp Suicide

To the Editor:

On the "Causes and Cures" page of the December number of Product Engineering, there is shown a method for redesigning a high-wattage street lamp reflector by putting flutes in it to reduce "lamp suicide." Premature lamp failures in inclosed street lighting luminaires, caused by excessive heat, can also be minimized by correct lamp location and special proportioning of reflector contours. Primary cause of the damaging temperatures which cause lamp failures is the re-direction of the lamp's radiated heat rays, by the reflector, right back into the base of the lamp.

In designing their Reflectolux and Refractolux street lighting units, Westinghouse engineers discovered that development of this high temperature was due not only to re-direction of heat rays back into the lamp, but that the location of the lamp in the reflector itself also had quite an effect.

Experiments disclosed that by locating the lamp light-center deeper inside the reflector, and then designing the reflector contour so that the re-directed heat rays came at a point below the lamp press, temperatures inside the lamp, lamp press, and lamp base fell to lower values than with any other method tried. Such a design is illustrated in the diagram.

In a series of tests of various reflectors using 10,000 lumen PS-40 lamps in the base-up position, temperature rise of the socket was 130 deg. C., or from 4.6 to 13.1 percent lower than in reflectors not designed to re-direct radiation away from the lamp press. Temperature rise in tests such as these is determined by means of thermocouples placed in the lamp press and socket.

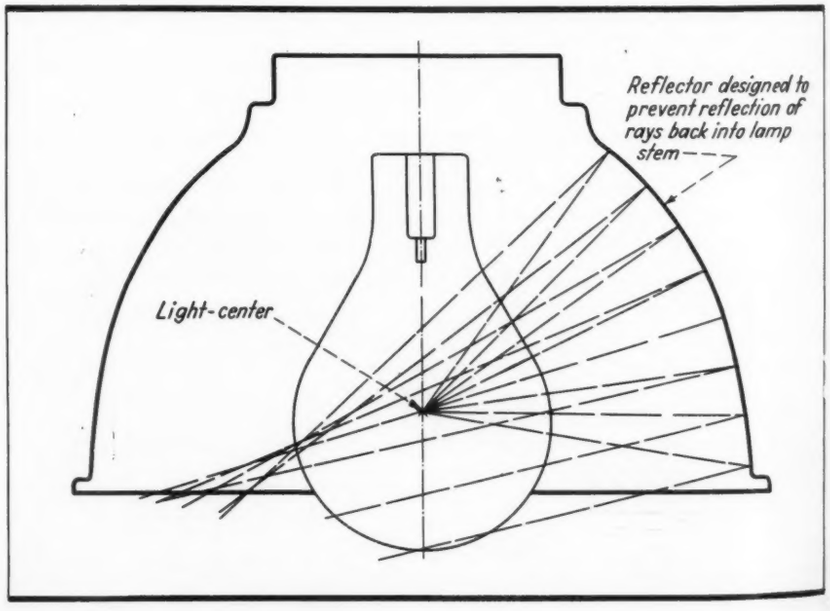
—S. B. Kraut
Street Lighting Engineer, Westinghouse Electric & Mfg. Company

Effective Bearing Loads For Harmonic Cycles

To the Editor:

Product Engineering published on page 138 of the March number a discussion by J. J. Pesqueira explaining a method for finding the mean effective load on a ball or roller bearing when the load cycle is known. The method explained by Mr. Pesqueira is general and can be applied to any load cycle. The special case of harmonically varying load is commonly encountered and deserves special consideration.

The load is proportional to sin θ where θ is the angle of the cycle expressed in radians. Also assume that the bearing life varies inversely as the cube of the bearing load. Then the effective load is the root-mean-cube



value of the load cycle. Therefore

$$\left(\frac{P_e}{P_m}\right)^3 = \frac{1}{\pi}\int_0^{\pi} \sin^3\theta \, d\theta = \frac{4}{3\pi}$$

$$\frac{P_e}{P_m} = \sqrt[3]{\frac{4}{3\pi}} = 0.7515$$

Where P_e = effective load; P_m = maximum load.

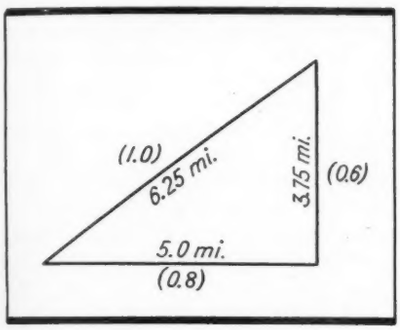
The effective load is approximately three quarters of the maximum load.

—N. D. SANDERS
Hampton, Va.

Engineer Rowing Ashore Could Be More Efficient

To the Editor:

The method used by the "efficient engineer" ("Can You Work This One?", January) to calculate the quickest way



to his cottage is quite suitable, but I feel that his efficiency rating should be reduced. Perhaps if he had spent less time on derivatives and more time on arithmetic he would have arrived at his cottage a few seconds sooner.

The time required as published in the current issue of your good periodical is 5.34 hours.

The triangle in the diagram represents relative rates where 1 hr. is required by boat, 0.6 is required to walk the same distance. The side adjacent (0.8 relative rate) is only for determining distances on other sides of triangle. Therefore,

distance by boat	= 6.25 mi.
time by boat	= 2.0833 hr.
distance by beach	= 20 − 3.75
time to walk	= 3.25 hr.
Total time	= 5.3333 hr.

—R. B. TUTTLE
Mt. Vernon, Ill.

Supplements Data On Molded Brake Linings

To the Editor:

The article entitled "Friction Materials" published in your January 1941 number was very interesting and I enjoyed it very much. However, there are a few statements which I believe require further clarification.

In the discussion on molded brake linings, the following sentences appear on page 6:

"Rigid molded materials being formed under pressure and the binder set to a rigid and infusible condition usually are more resistant to heat and more constant in frictional values over a relatively wide range of temperature than are the flexible types. Rigid molded linings are also less compressible and so require less frequent adjustment when high pressures are employed.

"Molded linings in roll form are usually made with an integral backing material to increase their mechanical strength, and are suitable for either internal expanding or external contracting wrap band or shoe type brakes. These linings have the friction facing on one side only and have to be curved to suit design with the backing on the proper surface, that is, internally for expanding and externally for contracting brakes. Coefficients of friction range from 0.20 to 0.50. The linings are highly resistant to wear and are suitable for rubbing speeds up to 5,000 ft. per min., drum temperatures up to 500 deg. F. constant service, and up to 100 lb. per sq. in. brake pressure. Flexibility depends on incompletely curing the organic binder and this type is more inclined to fade at high temperatures."

For years now the American Brakeblok Division of The American Brake Shoe and Foundry Company has made a flexible molded brake lining which is highly resistant to heat and in which the flexibility is not dependent upon incomplete curing of the organic binder. Actually, the flexibility of this American Brakeblok roll lining is obtained through the use of a highly heat resistant flexible bonding material. The bond itself is completely cured and the flexibility of the material depends upon the use of this heat resistant type of bond rather than upon incomplete curing of the organic binder.

Under extremely high unit pressures this flexible brake lining does compress slightly more than our rigid brake linings. However, under the pressure conditions normally encountered in brake operation this difference in compressibility is so small as to be negligible, and frequency of adjustment due to compression would be no greater with this type of flexible lining than with a rigid friction material. In fact, the rate of wear would be a much more important factor in causing frequent brake adjustments than would compressibility of the material. Numerous tests which we have made have confirmed the fact that there are soft or flexible brake linings which have better wear factors than some hard or rigid linings. Actually there can be no set rules regarding the compressibility, heat resistance, frictional characteristics or wear factors of flexible or rigid linings, considering either as a class.

I hope that the foregoing discussion will supplement and clarify the helpful information which I know all your readers obtained from your article on "Friction Materials."

—W. A. BLUME, *President*
American Brakeblok Div.
American Brake Shoe & Foundry Co.

Can You Work This One?

This month's problem—

Did Tilden Win This One?

A tennis tournament was held with Richards, Cochet, Hunter, Tilden and Johnson participating. From the following facts, determine the order in which the men finished:

1. Cochet had once defeated the winner in ping pong.
2. The man who finished in fourth place left immediately after his match. He went to Chicago, where he heard over the radio a play-by-play description of the finals the next day.
3. Previous to the tournament, the winner had never seen the man who finished fifth.
4. Richards and Johnson frequently played squash together.
5. Hunter surprised the gallery by beating Johnson.
6. The winner had lunch with Hunter the day of the finals, and was introduced there for the first time to his opponent of the afternoon by Hunter.

Solution to March Problem —

Problem of Thrift

A man, to train his three sons in thrift, opened a bank account for each with a deposit of $1 in each account. For the first dollar deposited by each son he added $2 to that son's account; for the second dollar he added $4, and so on. Total of the three deposits is now $200, of which the sons deposited $5, $7, and $9, and to which the father added $31, $57, and $91, respectively.

News and Summaries

The Aluminum Situation

CURRENT INCREASES in aluminum production (from 42,675,000 lb. in March to an estimated 58,000,000 lb. in December to an estimated 75-80,000,000 lb. by July, 1942) are astounding to market men. Aluminum producers are assuring their customers that these increases will alleviate present shortages. But OPM apparently still is a bit worried and, although slightly loosening up on priorities, is studying ways to replace the metal. In the middle of April an advisory group of the National Academy of Sciences recommended to OPM:

1. Toll fabrication of scrap.
2. Use of thinner gages.
3. Use of dross, sawdust, etc., in place of ingot for steel.
4. Use of more 3S (manganese alloy) in place of 2S (pure metal) for greater strength and thinner gages.
5. Regulations governing classification of scrap and designation of secondary alloys made from scrap.
6. Increased education.

Above and beyond these, the committee, cautioning that generalities with respect to substitutions are hazardous, and cautioning that such materials as brass, zinc, chrome-plating and nickel plating are themselves scarce, listed the following substitutions already in effect:

Aluminum foil—oiled paper, glassine, tin foil, lead foil, cellophane.

Bottle caps—paper, plastics, sheet steel protected by plastics or paint.

Cooking utensils—cast iron, sheet steel, glass, pottery.

Molding and trim—plastics.

Motor housings—cast iron, sheet steel, plastics.

Stove grids—cast iron.

Air blower wheels—plastics, welded.

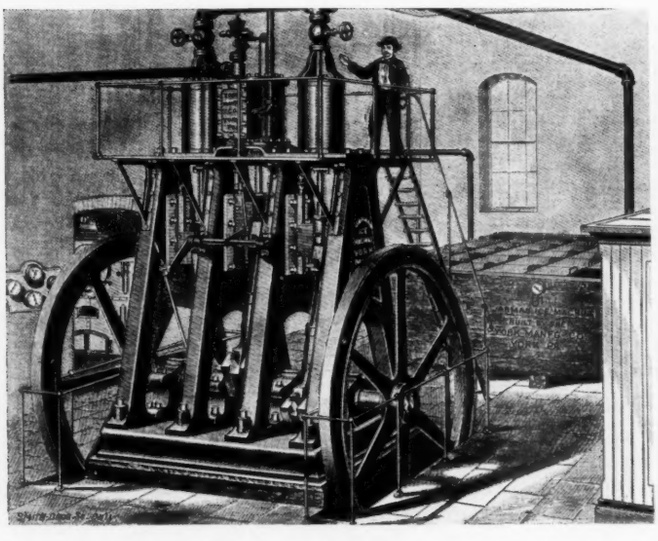
Washing machine agitators—plastics.

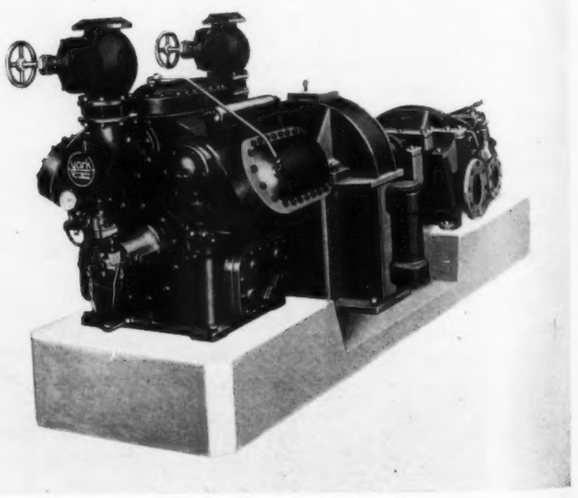
Other industries have been quick to suggest substitutions. The Porcelain Enamel Institute reports ice trays, cooling unit fronts and doors, vegetable pan fronts, and other refrigerator parts are now being made of porcelain enamel instead of aluminum. Chrome and aluminum trim in stoves can be replaced with attractive colored porcelain enamel.

Plastics are coming to the front as a replacement material for aluminum in extruded molding, rods, tubes, and sheet. The shortage of non-ferrous materials in Germany has led to much wider use of plastics there than here.

The immediate outlook is uncertain. Some authorities feel that increased production capacity will offset present commercial shortages, that government estimates of aluminum defense requirements are perhaps too high—in short, that there will be plenty of the material to go around within a few months. Others are fearful that some move such as the ordering of an additional 25,000 planes, each requiring an average of 2 tons of aluminum, may prolong the shortage. Sum of their advice to design engineers is to determine how aluminum can be replaced in their products, but not to put them into effect unless absolutely necessary, or unless advisable for reasons other than the shortage.

The long term outlook offers interesting speculations. It has been suggested that when the present emergency is over, the tremendously expanded aluminum production facilities will lead to considerable reductions in the price.





Jarman Ice Machine of 1885, York's first, offers striking contrast to the modern high-speed compressor of same capacity shown at right. This ponderous machine stood about 20 ft. high, was driven by a steam engine (center cylinder), turned over at about 50 r.p.m., and had a capacity of 100 tons per day. Its modern counterpart, York Ice Machinery Corporation's new "V/W" ammonia compressor, stands only 4 ft. high, occupies 1/60 the room space, and operates at speeds up to 1,050 r.p.m. The 6-cylinder compressor has water-jacketed heads and is driven by steam turbine (rear) through reduction gears. Elimination of vibration permits lighter foundations.

O. P. M. to Study Finishes Industry

SPECIFIC PROBLEMS of the protective coatings industry arising from the defense program will be analyzed by subcommittees now being set up by OPM. Members of the thirteen subcommittees will include representatives of the 1,000 manufacturers of protective coatings, technical experts from the 300 or 400 raw materials manufacturers, and the Government.

An adequate supply of proper finishes is vital to the defense program. Not only must warships and tanks be protected by paint against rust, but also the vast army cantonment and defense housing programs call for huge quantities of paint, such items as airplane bombs must be given a protective coating if they are to be serviceable, and even food cans call for protective coatings.

Many problems have developed in connection with supply. Tung oil and natural resins, vital ingredients in many paints, come from the interior of China, and war conditions have cut shipments heavily. Supply of many other raw materials, from familiar ones like aluminum to uncommon ones like phthalic anhydride, is not too abundant.

Prizes Established For Welding Aircraft Steels

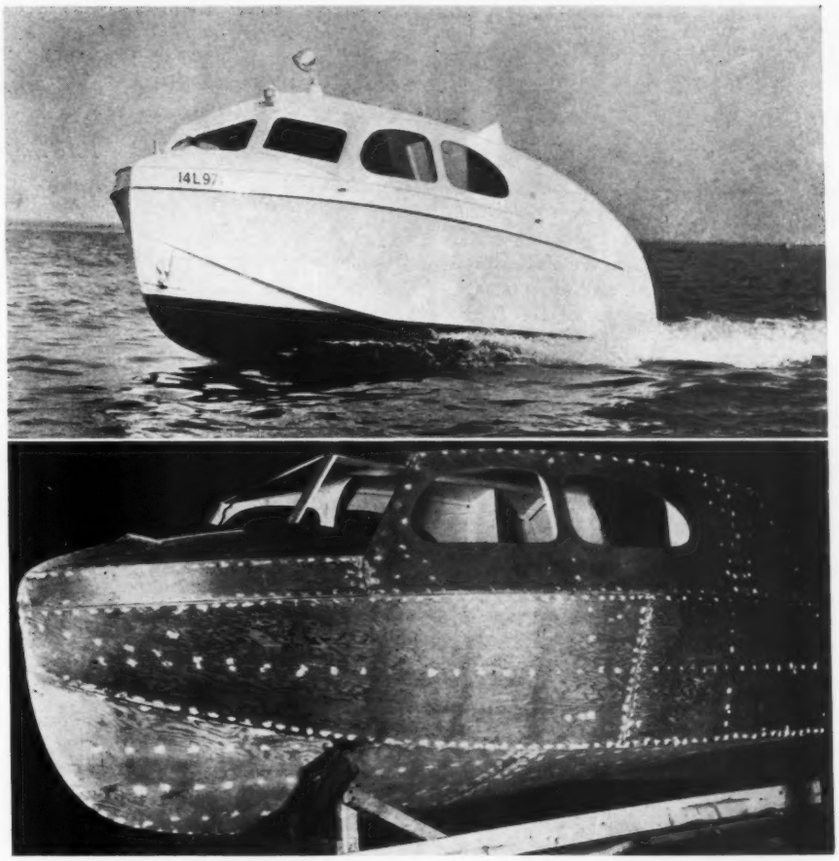
PRIZES TO BE AWARDED by the American Welding Society at its annual meeting during October, 1941, for papers advancing the art of welding aircraft steels, including tubing and other steel parts for tubular assemblies, have been established by the Summerill Tubing Company. One prize is of $300, one is of $200, and others are of $100 each.

Papers may be written on any type of welding which can be used in aircraft production with steels such as 1025, X4130, X4135, X4340. They may cover joint design, fabrication, or laboratory investigations. Papers must be submitted not later than August 18 to the American Welding Society.

60,000 Technical Men Take Advanced Training

UNDER THE AUSPICES of the United States Commissioner of Education more than 60,000 engineers, chemists and other technically-trained men now employed in industry are taking advanced training through extension courses conducted jointly by engineering schools and industries in all sections of the country. More than 1,000 courses are being offered by 127 institutions.

This program has developed from the original plan of giving special courses to students in the engineering schools (P.E., Dec. 1940, p. 575). Some 5,000 are taking such student courses, but the trend is toward almost the entire effort being made on in-industry training of men who have graduated and who have been employed for some time in lines similar to those for which there is a great need for special knowledge and experience.



Resin-bonded plywood, Plexiglas windows, a hull designed on same principles as the Boeing Clipper, built-in spray deflectors such as are used on Navy torpedo boats, and "flying stern" that rides out of the water at high speeds all contribute to the remarkable performance and economy of this new Doane "Speed-Ship." Hull is constructed of large sheets of resin-bonded plywood on the stressed-skin principle, requiring only one-fifth as many seams and one-third the weight of the conventional hull. Area of windows is 20 sq.ft., hence lighter Plexiglas instead of plate glass.

Manganese Shortage Growing More Acute

RECENT DEVELOPMENTS in the war have intensified the nation's problem of obtaining adequate supplies of essential manganese for steel production, Langbourne M. Williams, Jr., president of Cuban-American Manganese Corporation, declared in the current issue of *Chemical Industries*. Shipments no longer come from Russia, which for years has been our leading source of ferro-grade manganese ore. Shipping rates from other sources—India, the Gold Coast, South Africa, Brazil—have increased strikingly, in some instances as much as 500 percent since the war began. Domestic deposits produce less than 10 percent of our requirements.

The increasing difficulty of getting ships in order to bring foreign ore to our port has in recent months made the manganese problem one of active concern. An estimated 14 lb. of metallic manganese are used in the production of every ton of steel, and there is no commercial substitute for manganese in steel, stated Mr. Williams.

ENGINEERING IN WASHINGTON

McGRAW-HILL WASHINGTON BUREAU, *PAUL WOOTON, CHIEF*

Substitution of plastic for aluminum, magnesium, zinc and other strategic metals has been urged by E. R. Stettinius, director of priorities of OPM. Stettinius advised manufacturers to get in touch with Dr. Gordon Kline of the Plastics Section of the Bureau of Standards. The Bureau has noted a sharp increase in inquiries since the Stettinius notice. Interest covers not only those using the metals mentioned, but the whole field of manufacturing. Great progress in the plastics field may result, but probably there will no quick and startling developments. Experts remind readers that it was thousands of years before men made an engine out of iron, and it may be a long time before large stress members parts to meet peculiar requirements will be perfected. The Bureau has a project assigned by the National Advisory Committee for Aeronautics to develop special plastic materials for aircraft. This should be regarded as a long-term program, not likely to affect immediate defense needs.

Two spark plugs using ceramic insulation instead of mica have been approved by the Air Corps. There is a potential shortage of mica. Tests are showing that the flaky surface of mica tends to break down under overload conditions, and to short. Maintenance is said to be simpler on ceramic plugs, too. They are manufactured by A. C. and Champion.

Diamond dies for wire drawing are now under OPM priorities to defense industry. Applications for the diamonds are handled by the Defense Supplies Corporation. Not more than 6,000 are available. This will raise allocation problems. The dies are obtained from the British purchasing commission, and are used for drawing fine wire, such as aluminum, tungsten, and copper for delicate electromagnetic equipment. A hole is drilled through the center of the stone with a needle and abrasive diamond dust. Then the diamonds are mounted in steel discs which are held in the drawing machine. Wire is cold-drawn through a series of progressively smaller dies to the size wanted. There is no abrasion; only compression and stretching. The wire being drawn is lubricated. The smallest practical gage is about 0.0003 inch.

Rubber under pressure, functioning as female dies in forming and cutting sheet metal airplane parts, is speeding production in many plants. The National Advisory Committee for Aeronautics says the process has been in use for some years but is taking on more importance. Douglas Aircraft is using a system perfected by Henry Guerin, which is also licensed throughout the industry. Four large hydro-presses at Douglas are approaching an output of 44,000 units in 24 hours. Many of the parts are as large as most automotive pressed parts. Pressures range up to 10,000,000 lb. per sq.in. The company has presses of 5,000, 2,500, 650, and 250-ton capacity. The two larger ones use the Guerin process, while the others are of a special triple-acting type for deep-forming.

4,000 searchlight trailers are being built for the War Department at a cost of $4,465,040. Lights previously were moved on trucks requiring 11 ft. headspace which frequently caused detours. New trailers are 4-wheel tandem tilting type, taking only 9 ft. headspace. It has a new form of "knee action" which makes it ride evenly on rough ground. Wheels are mounted at each end of a walking beam, pivoted at the center. This arrangement, combined with a shock-absorbing mechanism, removes most of the bounce which could damage delicate parts of the lamp. The vehicle has a smooth underside which prevents hanging on obstructions. It has a collapsible dolly wheel for maneuvering, and four screw-jacks for leveling for operation. The trailers also carry the power plants.

Quick Watson! the jewels! The ramifications of details in a big defense push are almost beyond imagination. The other day the Government had to go out, with $45,000, and expand a plant that makes jewels to put in instruments to put in airplanes.

Amphibians for the marines are being ordered by the Navy. Contract has been awarded to Donald Roebling, Clearwater, Florida, for 200 units at a cost of $3,240,000. Mr. Roebling developed his craft as a duckboat. Since Navy took it over it has been thoroughly tested and improved. On land the tank has a speed of 25 m.p.h. and in water, 8 m.p.h. Traction is caterpillar, with extra wide cleats which furnish propulsion in the water. It operates on swamp lands and bogs where land and water alternate, which could not be covered by either boats or tractors. The craft weighs 8,000 lb., is 50 ft. long, and carries 50 men.

A metal container of a new type has been developed by the Quartermaster Corps for handling water, oil and gasoline. The container is being further developed, with insulation for food and drink. It is made of hot dipped galvanized steel; is rectangular in shape, about 6 inches thick, 13 inches wide, and 18 inches high. The rectangular shape facilitates storage. The container can be easily opened by hand. A vent pipe attached inside the spout provides an even flow of liquid. Orders have been placed for nearly 700,000.

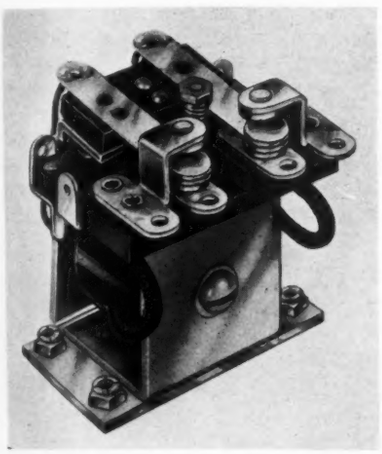
Controllable pitch marine propellers have been used on small speedy craft for some time. John Hays Hammond, Jr., inventor, and George Colley, naval architect, are seeking patents on such a wheel for general shipping use. Asked their opinion of this project, federal ship design authorities said that the loss of thrust in tow boats, due to slow speed, is pronounced; in heavily laden vessels there is some loss; in most vessels under normal conditions the loss is not great. They said that if controllable pitch could be achieved with comparative simplicity of mechanism, reasonable cost and dependability, it would be valuable. The proposition was very interesting. Beyond that they would not commit themselves until they have more information.

Damaged water pipes and pollution of water in seven communities were investigated by the American Research Committee on Grounding. In no case did the investigators find that alternating current flowing on the pipe had done any damage to metal or water. In all cases, the cause of the complaint was something else: chemical composition of the water; impurity of the metal; galvanic action resulting from indiscriminate use of many different metals; difference in temperature; or some other reason.

New Materials and Parts

Aircraft Relay

Special line of precision-built aircraft relays are designed particularly for minimum size and weight as well as for maximum reliability and resistance to vibration. Coils are wound on heavy Bakelite bobbins to insure low leakage and high voltage breakdown. Thorough impregnation is made with high quality baking varnish. All units are built



around the dynamic balanced armature principle. Contact and coil lugs are located on top of the relays for easy accessibility. Fine silver is used as the standard contact material with special alloys available for particular applications. Coil specifications range from 5 to 220 volts a.c. and from 1 to 125 volts d.c. Allied Control Co., Inc., 227 Fulton St., New York, N. Y.

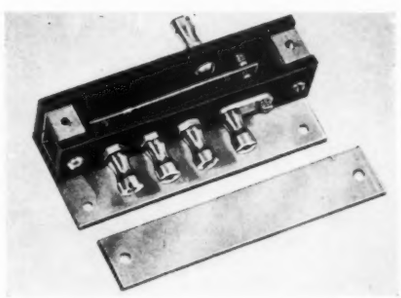
Metal-Clad Micro Switch

Roller arm metal-clad Micro Switch has a roller arm that is adjustable vertically through an arc of 225 deg. around its pivot pin, and also has a horizontal adjustment in eight positions 45 deg. apart. A ¾-in. dia. roller of hardened and ground steel, carried on an oilless bronze bearing, is suitable for slide or cam actuation. The arm is a lightweight aluminum die-casting. Zinc die-cast housing incorporates the small precision Bakelite Micro Switch as the switching element. The switch is rated at 1,200 watts up to 600 volts a.c. It is single pole only, with normally closed,

normally open, or double throw contact arrangements. Switch operates on only 2 to 4 deg. movement with provision for 20 to 30 deg. overtravel of the arm. Micro Switch Corp., Freeport, Ill.

Time Delay Switch

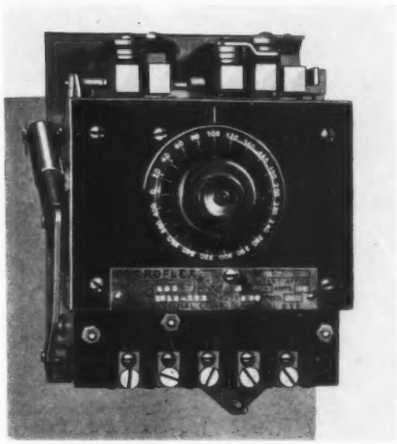
This new time-delay switch, which provides a predetermined time delay in electrical control, is designed for laboratory and industrial applications in conjunction with magnetic relays and other uses where temperature changes are fairly constant and where slight variations in timing are not detrimental.



Special compensating switches are provided where accurate delay is essential. Compact switches are provided with four terminals, two for the heater coil, two for the main circuit. These are adjustable within the time limits of 1 sec. to 5 min. Available in immediately or not immediately recycling types, normally open or normally closed models. Betts & Betts Corp., New York, N. Y.

Reset Counter

The Microflex counter, a relay for limiting the number of operations of a machine or part in a processing cycle, is fully automatic, requiring only electrical impulses to accomplish the counting operation. Dial settings remain fixed through any number of cycles, and need only be reset when the operator decides to change the number of operations of the machine. It is particularly well adapted for use in control circuits of automatic machine tools and process controls in which a contact is to be closed for a predetermined number of



operations. The counter automatically resets when the clutch coil is deenergized. The Microflex counter has an S.P.D.T. contact, one side of which is normally closed, and the other side normally open. Counting range is from 0 to 400 in steps of 1. Settings can be read at a glance. Available in all standard commercial voltages and frequencies. Eagle Signal Corp., Moline, Ill.

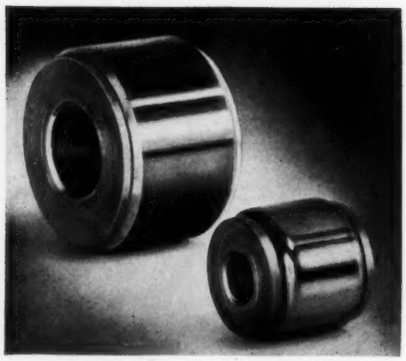
Torflex Bearings

Torflex bearings, rubber backed for insulation of noise, vibration, shock, are now carried in a number of stock sizes, representing the more popular of the sizes heretofore manufactured exclusively as specials. While standard as to diameters and in two torsional angles, even the standard sizes are available without extra cost or delay in shipping in optional or special lengths. This is made possible by the method of manu-

facture whereby the bearings are first made up in 4-ft. lengths and then cut to required lengths by high speed saws or in automatic screw machines. Range of the standard sizes, in diameters, is from $\frac{3}{16}$ in. to $\frac{3}{8}$ in. inside diameter. While most of the bearings are available with seamless steel inner and outer walls, a few of the most popular sizes to be used as rotative bearings are stocked with brass or graphited bronze inner walls. Harris Products Co., 5408 Commonwealth Ave., Detroit, Mich.

Needle Aircraft Bearings

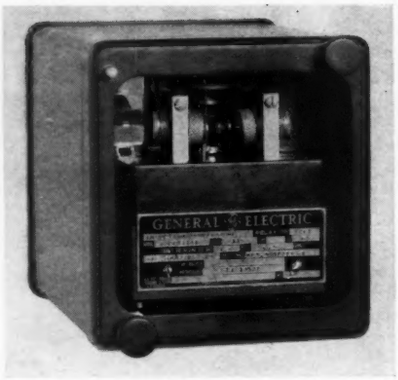
A number of new needle bearing assemblies, especially intended for the requirements of aircraft service, are made in two series, designated as the "AR" series for normal service and



"AT" series for heavy duty. Bearings in both series are provided with inner races and permanently attached end washers. All exposed surfaces are cadmium plated. Additional information and catalog are available on request. The Torrington Co., Torrington, Conn.

Temperature Relay

This improved temperature relay designated Type CFT has been developed for protection of machines and transformers against overloading. It embodies the induction-cylinder design, and supersedes the induction-disk relay. No change has been made in the circuit, and the improved relay still serves as the galvanometer in a Wheatstone bridge circuit comprising a resistance temperature detector in one arm and tapped bridge resistors in the other three arms. It is adjusted to close its low-temperature contact at 80 deg. C, and its high-temperature contact at 95 deg. C with a 30 deg. C ambient temperature. The balance point is midway between, as determined by the tap used on each of the three tapped bridge re-



sistors. The other taps permit adjustments, so that the balance point can be made higher or lower, and still keep range between the closing of the contacts. The relay is affected very little by ordinary variations in the voltage supplied to the bridge circuit. General Electric Co., Schenectady, N. Y.

Industrial Rectifier

These Electron industrial rectifiers are designed to supply d.c. from a.c. lines for the purpose of driving adjustable speed motors for machine tools, or for any other similar industrial power problem where d.c. is needed. These units are silent, cost less than rotating equipment, and are portable for units up to and including ½ hp., being equipped with plug connections.

Housed in ruggedly built, scientifically ventilated steel cases, properly fused. Case can be locked. Standard rectifiers are for use on 50 or 60 cycle lines, 110 or 440 volts a.c. input, in sizes from ¼ to 10 hp. Rectfiers for other voltages, frequencies or horsepower ratings can be furnished promptly on special order. Electron Equipment Corp., Palm Springs, Calif.

Flat Transmission Belt

A new method of combining 34-oz. duck with a special impregnating material provides improved grip and greater resilience in this new Octopus flat transmission belting. The belt runs at low tension to protect bearings and to provide longer service. The belting has a distinctive green color. Octopus requires no belt dressing, and is not recommended for oily drives. Belt rating data sheet and description is now available. Manheim Mfg. & Belting Co., Manheim, Pa.

Pressure Switch

Pressure switch No. 611-030, is designed for use on domestic water pump, pneumatic and oil systems, to control the stopping and starting of small

single- or polyphase a.c. motors on predetermined pressure settings. Working pressure ranges up to 80 lb. per sq.in. inclusive, and the differential between the "on" and "off" position may be specified and adjusted according to customers' requirements. Adjustments for various pressure ranges are easily made. Jefferson Electric Co., Bellwood, Ill.

Bearing Greases

Two new lines of lubricating greases for ball and roller bearings are available in a wide range of consistencies for any method of application or operating condition. One, designated Gulf Anti-Friction Grease, is recommended for heavy duty service. The other, Gulf Precision Grease, is recommended for lighter

duty and higher speeds. Both have a high melting point and are specially prepared for greater resistance to oxidation and separation. Gulf's newly developed method of compounding, employing special high pressure kettles and mixing methods, produces these greases with a relatively smooth, non-fibrous texture, which gives a true picture of their consistency. Consistencies recently recommended by the National Lubricating Grease Institute went into effect March 1, 1941. Gulf Oil Corp., Gulf Refining Co., Pittsburgh, Pa.

Cam Operated Controller

Extreme ease of operation with positive feel of all speed operations is obtained in these new cam-operated, mill duty electrical controllers by using an adjustable compression type of star-wheel spring. Available in two-speed, three-speed, and multi-speed types, the

controllers are ideal for mill auxiliaries, crane hoist, bridge, and trolley applications. Contacts are vertical, double break, silver-to-silver. Cam shaft operates on ball bearings sealed against dust. Easily accessible terminal board simplifies installation and service. A heavy cast case and cover, for either separate or bench board mounting, provide protection from dust and mechanical injury. Optional features include spring return, "off" position latch, two, three or five speeds. Cutler-Hammer, Inc., Milwaukee. Wis.

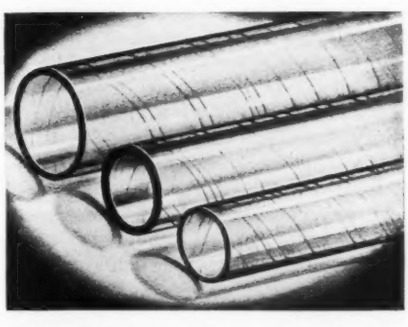
Hardness Tester

This new hardness tester, called "The Impressor" is for aluminum alloys or similar soft metals, plastics, hard rubber, and the like. It will permit testing the hardness of finished parts of almost any size or shape. It is small, light weight, and can be carried in the pocket for instant use. A slight pressure against the surface to be tested gives an instantaneous, direct reading on the plainly visible dial. No experience in the use of the instrument is necessary,

and its accuracy is not affected by the operator. The Impressor may be used in any position, vertically, horizontally, or even inverted, thus making it possible to check the hardness of vital points after parts have been fabricated in finished assemblies. Barber-Colman Co., Electrical Div., Rockford, Ill.

Transparent Tubing

This new transparent tubing is made by spirally wrapping acetate tape over a steel form of the required inside diameter, and using a new acetate cement for the adhesive to insure a solid non-separating wall. Being preformed, the tube will not shrink, eliminating one of the difficulties found in extruded acetate tubes. Due to its superior dielectric properties, it is recommended for many high frequency



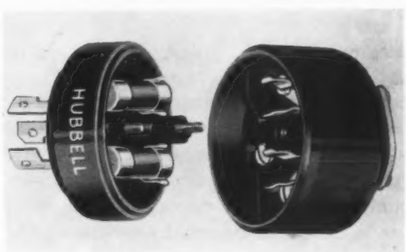
and electronic applications found in the radio and electrical industries, and for certain types and applications of low amperage cartridge fuses. Being highly transparent, not easily broken, and having a low moisture absorption rate, it has been suggested for liquid gages where temperatures do not exceed 180 deg. F. Precision Paper Tube Co., 2033 Charleston St., Chicago, Ill.

Varnished Fiber Glass

Glass cloth, impregnated and coated with heat resisting varnishes developed especially for this purpose, is now available for use as an insulation by manufacturers of electric motors, generators and similar equipment. The varnish coating over the woven glass cloth greatly increases its resistance to abrasion, impact, increases its overall mechanical strength, and results in a material with a higher insulation resistance and heat resistance than other flexible insulations. Standard thicknesses range from 0.005 in. to 0.012 in.; standard widths are available up to 36 in. Colors are black or yellow. Special combinations, varying as to thickness, coating, tightness of weave, filament size, and varnish, can also be developed. Irvington Varnish & Insulator Co., 24 Argyle Terrace, Irvington, N. J.

Fusible Cap

Fuse protection for the individual electrical tool or machine can be provided with this new three-wire fusible cap. The machine may be fused to its exact capacity, with no dependence upon the line fuse. This also protects a small motor from overload. Body is



made of special impact-resisting Bakelite, and exposed metal parts are cadmium plated to resist corrosion. Cap is made for use with 20 amp., three-wire Twist-Lock receptacles and connectors, and is available in capacities from 0 to 15 amp., 250 volts. Harvey Hubbell, Inc., Bridgeport, Conn.

Koroseal Paint

Developed to protect metal surfaces against chemical reactions and recommended for service where extremely corrosive conditions disqualify any other kind of paint or coating, the base of this new paint is Koroseal. When thoroughly dry, the paint is extremely resistant to the action of fumes and vapors from acids, alkalies and salts at room temperatures or slightly above. It resists all acids except concentrated formic and acetic, and is not affected

by brass, chrome, nickel, cadmium, zinc, copper, silver or tin plating solutions. The new paint is liquid at room temperatures and requires no heating before application. At ordinary temperatures it can be either sprayed or brushed, and can be thinned with either brush or sprayed thinners when necessary. It is made only in semi-glossy black. The B. F. Goodrich Co., Akron, Ohio.

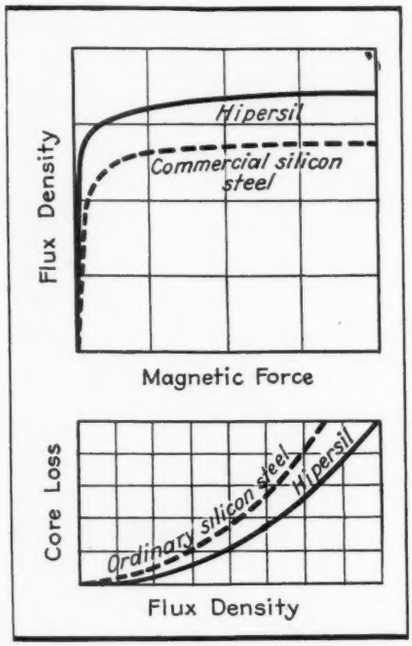
Small Mercury Switch

This new low-price all-metal mercury switch, called the "Tipit" switch, has a capacity of ½ amp. at 24 volts, and 4 amp. at 6 volts. The Tipit switch mounts in a small holder with screw stakes for terminal connection. It oper-

ates on a tilt of 20 deg. or more. It is used by automobile manufacturers as a convenient switch to turn the rear trunk light on when the lid is lifted and off when the lid is lowered. Other uses are for under-the-hood illumination, glove compartment lighting, advertising displays, and even toys and novelties. Also ideal for use in explosive atmospheres and on electrical machinery. Durakool, Inc., 1010 N. Main St., Elkhart, Ind.

Improved Magnetic Steel

This improved magnetic silicon steel, known as "Hypersil", has one-third greater flux-carrying capacity than the best conventional silicon steel. In carrying this increased flux it requires no more magnetizing force and its losses will be no greater. Furthermore, its magnetostriction, or sound producing property, is reduced so that increased magnetic flux causes no increase in sound level. In its first application it has reduced size and weight of distribution transformers as much as 25 percent, and has facilitated reduction of copper losses by 10 percent, thereby increasing short-time overload capacity. Curves of flux density plotted against magnetizing



force and core loss plotted against flux density are shown. The improved magnetic properties of Hypersil are produced by a special heat-treatment technique which rearranges the steel crystals in orderly patterns. The development was made jointly by American Rolling Mills Company and Westinghouse, and is now used exclusively by Westinghouse. For further details write Westinghouse Electric & Mfg. Co., East Pittsburgh, Pa.

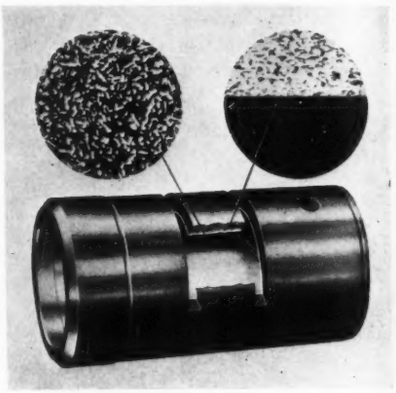
Capacitor Motor

This new Tri-Clad capacitor motor features modern appearance, and better mechanical and electrical protection as well as protection against operating wear and tear. It can be obtained with either ball-bearing or sleeve-bearing construction, and is available in two types, Type KC and Type KCJ. Type KC is designed particularly for applications requiring moderate starting torques such as fans, blowers, and centrifugal pumps. Type KCJ is designed for compressors, loaded conveyors, reciprocating pumps, and any other applications requiring high starting

torque. Capacitors are mounted inside the end shield in the normal-torque motor; in the high-starting torque motor capacitors are mounted in a compact case on top. In both of its forms, the capacitor motor incorporates all of the protective features of the Tri-Clad line. General Electric Co., Schenectady, N. Y.

Babbitt Lined Bearing

These new "Fusion Bonded" steel backed, babbitt lined bearings have a definite junction between opposite metals which prevents flexing and ultimate cracking of the bearing lining, and are claimed to have higher load carrying



capacity at elevated temperatures. The lining metal has inherent characteristics that produce a uniform structure, has good conformability, embedability, nonscoring qualities and retains hardness values. Present sizes are limited to inside diameters ranging from 1.125 to 4.00 in. Curtis Development & Mfg. Co., 3266 N. 33rd St., Milwaukee, Wis.

Phenolic Molding Compound

A new general-purpose phenolic molding compound, known as Durez 775 Black, has been developed to make available a material which would have a wider range of application than existing general-purpose materials. Among the improvements over present compounds of this type are: lower water absorption, slightly higher flexural and tensile strength, heat resistance of 400 deg. F. It is also stated that Durez 775 Black has excellent molding characteristics, fast cure, and will deliver a smoother, more lustrous finish than the average general-purpose material. Durez Plastics & Chemicals, Inc., Walck Road, North Tonawanda, N. Y.

Manufacturers' Publications

Materials

BEARING BRONZE—Johnson Bronze Co., New Castle, Pa. Catalog 410, 76 pages, 8½x11 in. Complete catalog listing of Johnson stock bushings, bearings, bar bronze, and babbitt. Some engineering data is included.

CELLULOSE ACETATE—Hercules Powder Co., Wilmington, Del. Bulletin 364-B, 28 pages, 8½x11 in. Gives data on properties, application and formulation of cellulose acetate in the plastic, lacquer, film, foil, paper, electrical, automobile and other fields.

INSULATION MATERIALS — Continental-Diamond Fibre Co., Newark, Del. Bulletin 8 pages, 8½x11 in. This well-illustrated booklet contains much selection data for laminated plastics, vulcanized fibre, mica, and other insulating materials.

MEEHANITE—Meehanite Research Institute of America, Inc., Pittsburgh, Pa. Bulletin 12, 8 pages, 4½x11 in. This folding type brochure illustrates and describes 12 industrial applications of different types of Meehanite castings.

SYNTHETIC RUBBER—Hydrocarbon Chemical & Rubber Co., Akron, Ohio. Bulletin, 8 pages, 8½x11 in. This non-technical booklet describes the physical properties of Hycar synthetic rubber and compares it with natural and other synthetic rubbers.

Mechanical Parts

BEARINGS—Clawson & Bals, Inc., Chicago, Ill. "Getting the Most from Engine Bearings," 32 pages, 8½x10½ in. This comprehensive booklet has been prepared as a manual data on engine bearings. Included are the causes and the diagnoses of most common bearing failures.

BRAKE AND CLUTCH FACING—S. K. Wellman Co., 1381 E. 49th St., Cleveland, Ohio. Bulletin, 40 pages, 8½x11 in. Pictorially shows the wide range of applications where Velvetouch, a friction material made from sintered powdered metals welded to steel, can be installed in friction clutches or brakes.

DIESEL ENGINES—Caterpillar Tractor Co., Peoria, Ill. Bulletin 5850, 48 pages, 8x10½ in. Discussion of diesel engine design with profuse use of cut-away photographs, treating each part of the engine separately. Horsepower ratings, complete specifications, and performance charts are included.

MINIATURE BALL BEARINGS—Landis & Gyr, Inc., 104 Fifth Ave., New York, N. Y. Bulletin 3, 6 pages, 8½x11 in. Illustrates, tabulates dimensions, and gives load capacity and speed ratings of miniature ball bearings available in sizes ranging from 1.5 mm. up to 20 mm. outside diameter.

SPEED REDUCER—The American Pulley Co., 4200 Wissahickon Ave., Philadelphia, Pa. Catalog R-41, 12 pages, 8½x11 in. Describes the new American Reduction Drive, and illustrates several typical applications. Data on how to select the correct drive are included.

SPEED REDUCERS—Winfield H. Smith, Inc., Springville, Erie Co., N. Y. Catalog No. 141, 36 pages, 8½x11 in. Complete catalog descriptions of all types of gear and differential speed reducers covering the range from fractional to 100 hp. with ratios from 2:1 to 20,000:1.

TANK SUPPORTS—Templeton Kenly & Co., 1020 So. Central Ave., Chicago, Ill. Engineering Data Form Tank-41, 8 pages, 8½x11 in. Describes Simplex adjustable tank supports and saddles for mounting wood or metal vessels of all types. Methods of fastening and welding are shown.

TRANSMISSIONS—Graham Transmissions, Inc., 2711 N. 13th St., Milwaukee, Wis. Manual 501, 8½x11 in. Principles of metallic friction, with explanatory curves and diagrams are set forth. The manual also summarizes applications of various types of variable speed transmission, and states when and when not to use a unit of the Graham type.

VALVES—Manning, Maxwell & Moore, Inc., Bridgeport, Conn., Bulletin 4-4500, 8 pages, 8½x11 in. Describes the application of Superfinish to the manufacture of the "500 Brinell" bronze valves.

V-BELTS—American Pulley Co., Philadelphia, Pa. Catalog W-41, 40 pages, 8½x 11 in. Contains complete engineering data for the design and selection of American "Wedgbelt" V-belt drives.

Electrical Parts

ADJUSTABLE SPEED DRIVE—Reliance Electric & Engineering Co., Ivanhoe Rd., Cleveland, Ohio. Bulletin 310, 8 pages, 8½x11 in. Describes the new all-electric adjustable-speed drives for a.c. circuits. These units are now available in horizontal instead of vertical mountings in the smaller sizes.

MAGNETIC STARTERS — The Trumbull Electric Mfg. Co., Plainville, Conn. Circular 330, 4 pages, 8½x10½ in. Illustrates and describes the new line of combination magnetic starters, now available in sizes No. 1, No. 2 and No. 3, which include a motor circuit switch, disconnect and magnetic starter in one cabinet.

MICRO SWITCHES—Micro Switch Corp., Freeport, Ill. Form 52-1, 36 pages, 9½x12 in. Complete engineering data on precision snap-action switches, including dimensions, prices, operating characteristics, and information as to their use.

MOTORS—Allis-Chalmers Mfg. Co., Milwaukee, Wis. Leaflet B-6131, 4 pages, 8½ x11 inches. Illustrates and describes Lo-Maintenance Type E, d.c. motors and generators for ratings to 200 hp. at 150 kw.

MOTOR CONTROLS—General Electric Co., Schenectady, N. Y. GEA-3531, 8 pages, 8x10½ in. "How to Select Control for D.C. Motors," is the title of this booklet, which includes a selection chart listing different G.E. control methods.

VOLTAGE REGULATOR — Westinghouse Electric & Mfg. Co., East Pittsburgh, Pa. Descriptive data 31-260, 19 pages, 8½x11 in. Describes the Silverstat voltage regulator for a.c. and d.c. generators. A data sheet lists all information required to make proper selection.

Instruments and Regulators

CONTROL EQUIPMENT—Wheelco Instrument Co., 1929-33 S. Halsted St., Chicago, Ill. Bulletin V-4000, 12 pages, 8½x11 in. Lists and briefly describes all Wheelco equipment for registering and controlling temperature, oil-fired equipment, and similar applications.

LIQUID LEVEL GAGES—The Bristol Co., Waterbury, Conn. Catalog 1015, 8½x11 in. Describes float, pressure bulb, pressure, air bubbler, counterpoise and differential pressure water and liquid level gages for all types of applications.

Engineering Department

ABRASER—Taber Instrument Co., N. Tonawanda, N. Y. Bulletin 4012, 6 pages, 7½x10½ in. Describes the new Research Model Taber Abraser. This new instrument incorporates many important improvements over previous models, all of which improvements are described in this bulletin.

BLUEPRINTER—C. F. Pease Co., 2601 W. Irving Park Rd., Chicago, Ill. Bulletin 22-16, 4 pages, 8½x11 in. Briefly describes and illustrates the new combination Model "22-16" blueprinting machine.

OFFICE EQUIPMENT—Angle Steel Stool Co., Plainwell, Mich. Bulletin 10-16-40, 4 pages, 8½x11 in. Illustrates many types of steel office equipment, many of which are adaptable to modern drafting rooms.

TRACING CLOTH—Arkwright Finishing Co., Turks Head Bldg., Providence, R. I. Catalog 1041, 6½x3¾ in. Contains all information necessary to pick out the proper type of tracing cloth. Samples of the five Arkwright tracing cloth products have been inserted.

Books and Bulletins

Anodic Oxidation Of Aluminum and Its Alloys

DR. A. JENNY, TRANSLATED BY WINIFRED LEWIS—*231 pages, 6¼x9 in., 107 illustrations, 36 tables. Published by Chemical Publishing Co., Inc., 234 King St., Brooklyn, N. Y. Price $6.50.*

Technically and commercially anodic films rank in importance far ahead of chemically produced protective films for aluminum and its alloys. The number and extent of existing and newly erected plants for the production of anodic films are in themselves sufficient evidence.

It is the purpose of this book to present an up-to-date general survey of anodic oxidation of aluminum and its alloys for the technical reader who lacks time to study the scattered literature on the subject. An introductory study of electrochemical theory, without which the processes of anodic oxidation cannot be understood, is given.

The author, Dr. A. Jenny, was Director of Research of the Siemens and Halske A.-G. Electrochemical Laboratories, and a foremost authority on the subject. In translating from the German, Mr. Lewis supplemented the text with information gained from the British Aluminum Company.

Temperature—Its Measurement And Control

1,361 pages, 6½x9¼ in. Published by Reinhold Publishing Corp., 330 W. 42nd St., New York, N. Y. Price $11.

This book is the record of a "Symposium on Temperature—Its Measurement and Control in Science and Industry," held in New York in November, 1939, under the auspices of the American Institute of Physics. The symposium was one of which the Institute could be justly proud, and had as its purposes: (1) coordination of the treatment of the subject of temperature in the several branches of science and engineering, (2) review of fundamental principles and a recapitulation of recent progress, (3) accumulation of contributions for a comprehensive text to be published after the symposium, (4) emphasis on the importance of temperature as a branch of physics, (5) improvement of technical curricula through making the latest information available. Chapters of particular interest to product engineers are: Precision Thermometry, Automatic Temperature Regulation and Recording, General Engineering, Optical and Radiation Pyrometry, and Thermometric Metals and Alloys.

Pressworking of Metals

C. W. HINMAN—*443 pages, 6½x9¼ in., 423 illustrations. Published by McGraw-Hill Book Co., 330 W. 42nd St., New York, N. Y. Price $4.*

Although this book is prefaced as "a reference book illustrating and describing practical applications of the principles used in the design of punches and dies," and is aimed at the production engineer, it is an excellent comprehensive treatise of a subject which ought to be fully understood by the progressive product engineer. A study of up-to-date principles of pressworking may lead to the redesign of parts in order to do by stamping a job formerly done by some more expensive process, to reduce scrap strip, to simplify pressworking operations, or to affect other production savings.

Pressworking of metals, according to Mr. Hinman, involves three outstanding factors: the press, the work material, and the die. In reviewing each of these fundamentals, the author first describes up-to-date presses, what they can do, and their accessories. Workability of and specifications for ordering sheet materials are next discussed, and the remainder of the book deals with die designing and operations.

• • •

Aircraft Tubing Data

EDITED BY JOHN E. YOUNGER—*142 pages, 9½x11½ in. Published by Summerill Tubing Co., Bridgeport, Pa. Restricted complimentary distribution; price to others $1.50, $2 with ring binder.*

In this one volume are contained practically all the data of value to the engineer relating to aircraft tubing. The book is divided into four sections. The first section covers general information, such as its manufacture, tolerances, government specifications, what the tubing mill can supply. The second section covers fabrication methods, including much data and illustrations which have appeared in PRODUCT ENGINEERING. Discussions on welding are particularly noteworthy. Section three includes 62 pages of design charts, design data, tables of properties and formulas for aircraft tubing. Section four contains many commonly used reference tables. This book will undoubtedly become the "bible" of aircraft tubing design.

Handbook of Special Steels

124 pages, 5¾x7⅞ in. Published by Allegheny Ludlum Steel Corp., Pittsburgh, Pa.

All special steels, such as tool, stainless, electrical and carbon steels, manufactured by Allegheny Ludlum are covered in this book, which lists data covering grades, heat-treatment, annealing, electrical properties, strength and elasticity. Tables have been used throughout the handbook to facilitate quick reference. Some of these tables list typical applications of the special steels. Reference tables listed in the back of the book give complete dimension and weight data as well as many other tables commonly used in the design and machining of special steels.

Felts for Sealing And Lubricating Bearings

R. R. STEVENS—*42 pages, 5½x8½ in. Published by the Felters Co., Inc., 210 South St., Boston, Mass.*

Purpose of this booklet is to give a general explanation of the properties and design engineering of felt. Supplementing the text, many photographic illustrations and engineering drawings illustrate applications of felt seals in ball and roller bearings, journal boxes, machine shafts. Tables included in the booklet list the physical requirements, thickness tolerances and chemical requirements for the various S.A.E. specifications.

Bolts, Nuts and Screws

COMPILED BY A. E. R. PETERKA—*70 pages, 9x12 in. Published by Lamson & Sessions Co., Cleveland, Ohio. Price $1.*

The sum total of bolt and nut manufacturing experience and research cannot be found in the files of any one manufacturer, and would be difficult to find in text books or in technical libraries. But, over the past few years a number of articles have been published on the subject in technical magazines, many of them written by Mr. Peterka. Fourteen of the best of these technical articles have been compiled in this book.

Hot-Tinning

Publication 102, 28 pages, 8½x10½ in. Published by Battelle Memorial Institute, 505 King Ave., Columbus, Ohio.

This booklet gives a comprehensive account of the different processes used in the tinning of articles fabricated from mild and carbon steels, copper and copper alloys, cast iron and certain alloy steels. Various types of tinning baths and factors governing their choice are discussed. Operating temperatures, bath maintenance, brightness of coatings and retinning are covered.

BELT DRIVES WITH IDLER PULLEYS

Belt Length and Arc of Contact

E. N. KEMLER
Associate Professor, Purdue University

FORMULAS for calculating belt lengths and arcs of contact for drives using idler pulleys have not been reduced to a usable basis, and are not normally considered in the texts or reference books on the subject. This Reference Book Sheet gives a practical and simple method for determining these values for this type of drive with any degree of accuracy desired. For most purposes the formulas can be reduced to simple forms as given here.

Belt Length

The approximate formula normally used for the length L of an open belt drive is

$$L = 2C + \frac{\pi}{2}(D_1 + D_2) + \frac{(D_1 - D_2)^2}{4C}$$

and for crossed belt drive

$$L = 2C + \frac{\pi}{2}(D_1 + D_2) + \frac{(D_1 + D_2)^2}{4C}$$

Where D_1 and D_2 are the pulley diameters and C the distance between pulley centers.

The belt drive using an idler pulley is illustrated in Fig. 1. The length of the belt can, for purposes of derivation, be best broken up into the lengths of P, Q, R, S, Y, U, V and W. The section P is obviously half of the length of an open belt drive going around pulleys D_1 and D_2. The sections Q and Y are the 90-deg. wrap around each of the pulleys D_1 and D_2. The sections R and S are parts of crossed belt drives whose center distances are A and B respectively and whose lengths are found by taking one-half the value of that equation which does not include the wrap around the half pulley diameter of each pulley. The overlapping of these various sections is shown by U, V, and W, and these lengths must be subtracted. These segments can be calculated if the angles α and β are known. As in the case of the belt formulas the angles α and β can be expanded in terms of a series giving a simpler solution. From Fig. 1, $\sin \alpha = E/B$, or $\alpha = \sin^{-1} E/B$

The length W will be equal to $\left(\frac{D_2}{2} \times \alpha\right)$

The arc sin can be expanded in a series giving

$$\alpha = \sin^{-1}\frac{E}{B} = \frac{E}{B} + \left(\frac{E}{B}\right)^3 \times 1/6 + \left(\frac{E}{B}\right)^5 \times 3/40 + \ldots$$

If $E/B = 1/2$ or less, the second term will be 1/48 or less, so that for any normal values the error will be small. As an approximation $\alpha = E/B$. Similarly, $\beta = E/A$.

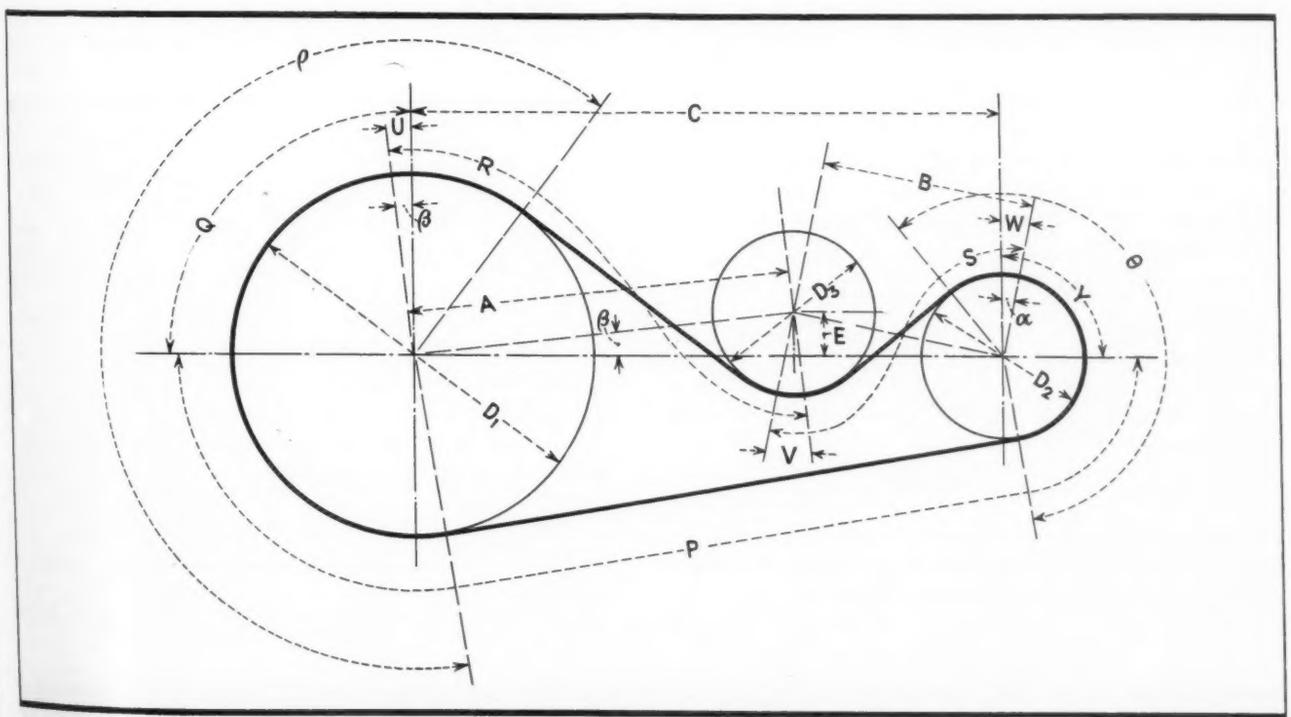


Fig. 1—Diagram of a belt drive with an idler pulley showing how the belt is divided into sections to facilitate computations

Applying the above formulas, we have the various lengths making up L to be as follows

$$P = \tfrac{1}{2}\left[2C + \frac{\pi}{2}(D_1 + D_2) + \frac{(D_1 - D_2)^2}{4C} \right]$$

$$Q = \frac{\pi}{4} D_1 \qquad Y = \frac{\pi}{4} D_2$$

$$R = \tfrac{1}{2}\left[2A + \frac{(D_1 + D_3)^2}{4A} \right] \qquad S = \tfrac{1}{2}\left[2B + \frac{(D_2 + D_3)^2}{4B} \right]$$

$$U = \frac{E}{A} \times \frac{D_1}{2} \qquad W = \frac{E}{B} \times \frac{D_2}{2} \qquad V = \frac{D_3}{2}\left[\frac{E}{A} + \frac{E}{B} \right]$$

Then $L = P + A + Y + R + S - U - V - W$

$$L = C + \frac{\pi}{4}(D_1 + D_2) + \frac{(D_1 - D_2)^2}{8C} + \frac{\pi}{4} D_1 + \frac{\pi}{4} D_2 + A$$

$$+ \frac{(D_1 + D_3)^2}{8A} + B + \frac{(D_2 + D_3)^2}{8B} - \frac{E}{A} \times \frac{D_1}{2}$$

$$- \frac{E}{B} \times \frac{D_2}{2} - \frac{D_3}{2}\left[\frac{E}{A} + \frac{E}{B} \right]$$

This reduces to the simple form

$$L = (C + A + B) + \frac{\pi}{2}(D_1 + D_2) + \frac{(D_1 - D_2)^2}{8C}$$

$$+ \frac{(D_1 + D_3)^2}{8A} + \frac{(D_2 + D_3)^2}{8B}$$

$$- \frac{E}{A}\left(\frac{D_1 + D_3}{2} \right) - \frac{E}{B}\left(\frac{D_2 + D_3}{2} \right)$$

If a more accurate formula is desired the more complete expansion of the arc sin can be substituted where E/A and E/B occur.

In the above equation E is considered positive if the center of the idler pulley falls above the center line of the other two pulleys as shown in Fig. 1. If it falls below the center line E becomes negative.

Example—As an example take a case where $C = 50$ in., $D_1 = 30$ in., $D_2 = 5$ in., $D_3 = 8$ in., $B = 12$ in., $A = 40$ in., and $E = -1$ in.

$$L = 50 + 40 + 12 + \frac{\pi}{2}(30 + 5) + \frac{(30 - 5)^2}{8 \times 50} + \frac{(30 + 8)^2}{8 \times 40}$$

$$+ \frac{(5 + 8)^2}{8 \times 12} - \frac{-1}{40}\left(\frac{30 + 8}{2} \right) - \frac{-1}{12}\left(\frac{5 + 8}{2} \right)$$

$$= 50 + 40 + 12 + 54.95 + 1.56 + 4.51 + 1.76 + 0.47 + 0.54$$

$$= 165.73 \text{ in.}$$

As a comparison the length of belt for an open drive around pulleys D_1 and D_2 is:

$$L = 2 \times 50 + \frac{\pi}{2}(30 + 5) + \frac{(30 - 5)^2}{4 \times 50} = 158.07$$

The increase in length as a result of the belt's going around the idler pulley is 165.73 − 158.07 = 7.63 in. In the example given the idler pulley will be very near to the lowest practical operating position. In this case, therefore, a takeup of 7.63 in. is possible.

Arc of Contact

PRODUCT ENGINEERING's Reference Book Sheet for February 1935, gave formulas for calculating the various angles which, when combined, give the arcs of contact. The arc of contact on the small or driving pulley and on the large or driven pulley are indicated in Fig. 1 by θ and ρ respectively. Using the same general methods as indicated in the above mentioned article and the notation shown in Fig. 1. these formulas can be written as

$$\theta = 180 - \frac{57.3\,(D_1 - D_2)}{2C} + 57.3\,\frac{(D_2 + D_3 - 2E)}{2B}$$

$$\rho = 180 + \frac{57.3\,(D_1 - D_2)}{2C} + 57.3\,\frac{(D_1 + D_3 - 2E)}{2A}$$

Example—Applying the data in the above case and using E as positive if above and negative if below the center line we have for this example:

$$\theta = 180 - \frac{57.3\,(30 - 5)}{2 \times 50} + \frac{57.3\,(5 + 8 - 2\,(-1))}{2 \times 40}$$

$$= 180 - 14.3 + 35.8 = 201.5 \text{ deg.}$$

$$\rho = 180 + \frac{57.3\,(30 - 5)}{2 \times 50} + \frac{57.3\,(30 + 8 - 2\,(-1))}{2 \times 40}$$

$$= 180 + 14.3 + 28.7 = 223 \text{ deg.}$$

The horsepower transmitted by a belt can be expressed in the form

$$Hp. = \frac{T_1 A V}{33{,}000}$$

where Hp = horsepower transmitted
T_1 = total tight side tension
V = velocity in ft. per min.

$$A = \frac{e^{\mu\theta} - 1}{e^{\mu\theta}}$$

where θ = angle of contact
μ = coefficient of friction

To illustrate how the arc of contact affects the horsepower, Fig. 2 has been drawn. If T_1 is fixed by the kind of belt specified the horsepower which can be transmitted will be determined by μ and θ. In the case worked above, A equals 0.64 for $\mu = 0.30$ and $\theta = 201.5$. If this had been an open belt drive the arc of contact would have been 151.4 deg. and for $\mu = 0.30$, A would be 0.535. In this case the use of the idler pulley would increase the horsepower by $\frac{0.64 - 0.535}{0.535}$ or approximately 20 percent.

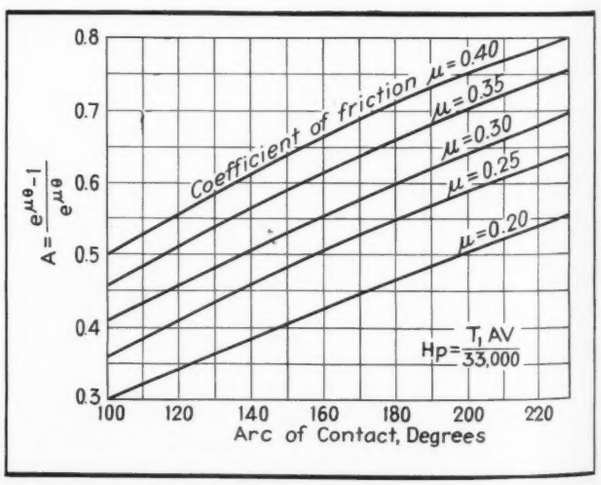


Fig. 2—Chart showing how hp. capacity of a belt drive can be increased by increasing arc of contact with an idler pulley